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**Engineers' Joint Council:** L. E. Seeley (ASHAE Representative).

**Exposition Rates:** E. N. McDonnell, *Chairman*; John E. Haines, L. N. Hunter.

**International Joint Committee on Psychrometric Data:** C. O. Mackey, Ithaca, N. Y., *Chairman*; B. A. Dmitrieff, New York, N. Y., *Secretary*; H. H. Bruce, London; G. A. Bull, London; C. S. Cragoe, Washington, D. C.; John A. Goff, Philadelphia; L. P. Harrison, Washington, D. C.; T. J. G. Henry, Toronto; B. H. Jennings, Evanston, Ill.; F. G. Keyes, Cambridge, Mass.; R. F. Legget, Ottawa; S. G. Rison, Washington, D. C.; P. A. Sheppard, London; J. L. York, Ann Arbor, Mich.

**Long-Range Planning Committee:** P. B. Gordon, *Chairman*; A. J. Hess, L. N. Hunter, John W. James, J. D. Kroeker, L. E. Seeley.

**NRC-National Academy of Sciences, Div. of Engrg.:** B. H. Jennings (ASHAE Representative).

**To Codify Council Policies:** J. H. Fox, *Chairman*; I. W. Cotton, A. W. Edwards, B. H. Spurlock, Jr.

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**Nominating:** B. L. Evans, St. Louis, *Chairman*; W. O. Stewart, Los Angeles, *Secretary*; F. W. Brundage, Kalamazoo; E. L. Crosby, Baltimore; W. J. Collins, Jr., Oklahoma City; J. D. Kroeker, Portland, Ore.; A. B. Newton, Wichita, Kans.; W. B. Pennock, Ottawa, Ont.; J. D. Slemmons, Columbus, Ohio; R. L. Stinard, New York; G. D. Winans, Detroit. *Alternates*: W. G. Hole, Montreal, Que.; G. A. Linskie, Dallas.

## Local Chapter Officers—1955

### Arizona

*Organized 1953*  
Headquarters, Phoenix

President.....W. A. Biddle  
Vice President.....J. P. Sorenson  
Secretary.....E. P. Maxwell  
Treasurer.....J. J. Vargo  
Board of Governors: V. J. Carns

### Arkansas

*Organized 1952*  
Headquarters, Little Rock

President.....Jack Franklin, J. L. Brown\*  
Vice President.....J. L. Brown, J. W. Thompson  
Secretary.....J. W. Thompson, H. H. Himstedt\*  
Treasurer.....H. H. Himstedt, K. A. Pettit\*  
Board of Governors: R. H. Combs, Luther  
Granderson, Jr., J. C. Lewis, C. H. Miller,  
Fred Tenny

### Atlanta

*Organized 1937*  
Headquarters, Atlanta, Ga.

President.....T. A. Barrow, Jr.  
Vice President.....F. W. Bull  
Secretary.....J. A. Marshall  
Treasurer.....J. S. Edgar  
Board of Governors: J. G. Croley, J. M. Lazenby

### Baltimore

*Organized 1949*  
Headquarters, Baltimore, Md.

President.....E. J. Morris  
Vice President.....W. G. Robertson, Jr.  
Secretary.....E. J. Dull  
Treasurer.....C. E. Rice, Jr.  
Board of Governors: R. E. Dressell, A. M. Norris,  
R. R. Poole, J. K. Wolford

### Baton Rouge

*Organized 1955*  
Headquarters, Baton Rouge, La.

President.....C. S. Woodruff  
Vice President.....J. F. Naylor, Jr.  
Secretary.....A. J. Mayers, Jr.  
Treasurer.....A. W. Holland  
Board of Governors: W. J. LaBlanc, W. I. Pol,  
F. W. Stone, Jr.

### Bluegrass

*Organized 1954*  
Headquarters, Louisville, Ky.

President.....W. M. Pace  
Vice President.....W. J. Killian  
Secretary.....R. B. Duggan  
Treasurer.....O. J. Hill  
Board of Governors: E. C. Arenz, A. E. Handmaker,  
F. E. Hanson, F. M. Kilroy, E. W. Neel, D. J.  
Rank, E. G. Zanone

### British Columbia

*Organized 1952*  
Headquarters, Vancouver, B. C., Canada

President.....Cornelius Van Boeyen  
Vice President.....D. W. Thomson  
Secretary.....A. C. Martin  
Treasurer.....R. D. Hale  
Board of Governors: D. M. Drake, D. B. Leaney

### Central New York

*Organized 1944*  
Headquarters, Syracuse

President.....L. A. Childs  
Vice President.....H. J. Morton  
Secretary.....Merle Weninger  
Treasurer.....K. T. Sprague  
Board of Governors: S. F. Gilman, E. L. Moyer

### Central Ohio

*Organized 1944*  
Headquarters, Columbus

President.....R. A. Wilson  
Vice President.....W. A. Schoonover  
Secretary.....T. R. Walker  
Treasurer.....R. C. Liebert  
Board of Governors: J. A. Guy, R. L. Keener, A. F.  
McGovern

### Cincinnati

*Organized 1932*  
Headquarters, Cincinnati, Ohio

President.....A. H. Gerdsen  
Vice President.....R. G. Anderson  
Secretary.....T. D. Reiley  
Treasurer.....C. P. Krantz  
Board of Governors: E. A. Sobolewski, F. W.  
Wilson

### Connecticut

*Organized 1940*  
Headquarters, New Haven

President.....E. J. Hoagland  
Vice President.....Walter Heywood  
Secretary.....J. D. Pierce  
Treasurer.....Fritz Honerkamp  
Board of Governors: R. B. Cahoon, C. L.  
L'Hommedieu, J. J. Morro

### Delta

*Organized 1939*  
Headquarters, New Orleans, La.

President.....W. B. Martin, Jr.  
Vice President.....H. N. Stall  
Secretary.....R. K. Goode  
Treasurer.....L. R. Maxwell  
Board of Governors: J. T. Knight, Jr., J. E.  
Leininger

\* Filled unexpired term.

## Local Chapter Officers—1955 (Continued)

### Empire State Capital

#### *Organized 1951*

Headquarters, Albany, N. Y.

President.....L. V. Appleby  
1st Vice President.....G. G. Davis  
2nd Vice President.....H. F. Kruger  
Secretary-Treasurer.....E. J. Mahoney  
Board of Governors: J. P. Hawn, R. D. Marshall,  
M. E. Waddell

### Golden Gate

#### *Organized 1937*

Headquarters, San Francisco, Calif.

President.....D. E. McLeod  
Vice President.....F. K. Crouch  
Secretary.....Herb Duncan  
Treasurer.....J. B. Smith  
Board of Governors: R. E. Conner, James  
McLachlin

### Illinois

#### *Organized 1906*

Headquarters, Chicago

President.....G. G. Freyder  
Vice President.....H. G. Gragg  
Secretary.....H. Kreisman  
Treasurer.....H. E. Anderson  
Board of Governors: J. G. Pennington, J. C. Scott,  
C. M. Vreuls

### Indiana

#### *Organized 1943*

Headquarters, Indianapolis

President.....James Jackson  
Vice President.....A. O. Roche, Jr.  
Secretary.....G. W. Vogel  
Treasurer.....W. W. Gear  
Board of Governors: W. F. Currise, P. O.  
Patterson, J. M. Teskoski

### Inland Empire

#### *Organized 1950*

Headquarters, Spokane, Wash.

President.....F. W. Jenkinson  
Vice President.....Max Tonn  
Secretary.....J. A. Doyle  
Treasurer.....R. D. Nevers  
Board of Governors: H. A. Bickel, R. B. Campbell,  
J. R. Morris

### Iowa

#### *Organized 1940*

Headquarters, Des Moines

President.....G. J. Kraai  
Vice President.....D. E. Schroeder  
Secretary-Treasurer.....W. A. Schworm  
Board of Governors: L. E. Gnade, W. E. Nanes,  
C. A. Wheeler

### Kansas

#### *Organized 1951*

Headquarters, Wichita

President.....A. B. Newton  
Vice President.....H. M. Skalla  
Secretary.....T. L. Roberts  
Treasurer.....A. P. Sullivan  
Board of Governors: R. F. Bauer, O. P. Bullock,  
R. L. Pennington, Charles Yoe

### Kansas City

#### *Organized 1917*

Headquarters, Kansas City, Mo.

President.....A. S. Hurt, Jr.  
Vice President.....R. M. Spencer  
Secretary.....S. C. McCann  
Treasurer.....J. E. Miller  
Board of Governors: J. R. DeRigne, F. K. Ladewig,  
R. B. Luhnnow, Jr.

### Manitoba

#### *Organized 1935*

Headquarters, Winnipeg, Man., Canada

President.....G. C. Davis  
Vice President.....A. J. McIntyre  
Secretary-Treasurer.....A. H. Millar  
Board of Governors: G. A. Blackwell, R. G.  
Cawker, G. T. Christie, R. M. Fraser, J. J.  
MacKenzie, A. K. Piercy, H. F. Randall, J. C.  
Stangl, J. R. Stephenson

### Massachusetts

#### *Organized 1912*

Headquarters, Boston

President.....G. D. Fife  
Vice President.....W. G. Martin, Jr.  
Secretary.....R. F. Curry  
Treasurer.....E. L. Blair  
Board of Governors: F. J. Butler, J. P. Hoar, B. H.  
Snow

### Memphis

#### *Organized 1944*

Headquarters, Memphis, Tenn.

President.....John Hilton, II  
Vice President.....W. L. Henson  
Secretary.....L. V. Eberle, Jr.  
Treasurer.....R. H. Bolding  
Board of Governors: W. L. Drake, R. E. Larkin,  
A. W. Shelby, H. H. Wilson

### Miami Valley

#### *Organized 1950*

Headquarters, Dayton, Ohio

President.....D. E. Tullis  
Vice President.....J. B. Rishel  
Secretary.....L. P. Brehm, Jr.  
Treasurer.....R. B. Walcott  
Board of Governors: W. D. Banks, C. J. Doudican,  
R. W. Kimmel, F. W. Spencer

## Local Chapter Officers—1955 (Continued)

### Michigan

*Organized 1916*  
Headquarters, Detroit

President.....D. S. Falk  
Vice President.....J. N. Livermore  
Secretary.....J. H. Spitzley  
Treasurer.....P. S. Hosman  
Board of Governors: C. J. Henstock, K. J. Wagoner, G. S. Whittaker

### Minnesota

*Organized 1918*  
Headquarters, Minneapolis

President.....J. S. Locke  
Vice President.....E. T. Erickson  
Secretary.....Fred Vogt  
Treasurer.....R. J. Ruth  
Board of Governors: A. B. Algren and J. F. Siegel

### Mississippi

*Organized 1953*  
Headquarters, Jackson

President.....J. E. Davis, Jr.  
Vice President.....J. W. Taylor  
Secretary.....L. H. Smith  
Treasurer.....J. W. Doggett, Jr.  
Board of Governors: F. L. Cooper and H. S. Thomas

### Montreal

*Organized 1936*  
Headquarters, Montreal, Que., Canada

President.....R. J. Ker  
Vice President.....S. R. Plamondon  
Secretary.....W. E. Jarvis  
Treasurer.....A. deBreyne  
Board of Governors: Robert Clapperton, W. G. Hole, J. G. LeFrancois, D. L. Lindsay, H. G. S. Murray

### Nebraska

*Organized 1940*  
Headquarters, Omaha

President.....T. E. Davis  
Vice President.....L. J. Paulsen  
Secretary.....C. A. Failor  
Treasurer.....M. F. Stober  
Board of Governors: H. A. Barnard, C. A. Goth, W. B. Howard

### New Mexico

*Organized 1954*  
Headquarters, Albuquerque

President.....H. F. Munn  
Vice President.....C. R. Wherritt  
Secretary.....J. K. James  
Treasurer.....R. W. Haines  
Board of Governors: F. H. Bridgers and R. G. Merryman

### New York

*Organized 1911*  
Headquarters, New York

President.....Albert Giannini  
Vice President.....J. B. Hewett  
Secretary.....C. H. Flink  
Treasurer.....W. O. Huebner  
Board of Governors: C. R. Hiers, W. O. Huebner, J. M. Levy, W. J. Olvany, R. L. Stinard

### North Jersey

*Organized 1952*  
Headquarters, Newark, N. J.

President.....C. H. Smith  
Vice President.....W. C. Kruse, Jr.  
Secretary.....M. C. Christesen  
Treasurer.....H. M. Patrick  
Board of Governors: L. G. Huggins, H. P. Morehouse, G. C. Norman, H. M. Patrick

### North Texas

*Organized 1938*  
Headquarters, Dallas

President.....M. W. Brown  
Vice President.....J. A. Ray  
Secretary.....H. G. Gregerson  
Treasurer.....G. H. Meffert  
Board of Governors: W. E. Frost, E. T. Keck, Jr., P. N. Vinther

### Northeastern Oklahoma

*Organized 1949*  
Headquarters, Tulsa

President.....R. W. Winget  
Vice President.....R. E. Pauling  
Secretary.....J. T. McKinney  
Treasurer.....J. D. Ryan  
Board of Governors: Fred Colbert, J. C. Netherton, Jr., J. N. Watt

### Northern Ohio

*Organized 1916*  
Headquarters, Cleveland

President.....L. C. Burkes  
Vice President.....H. R. Canoyer  
Secretary.....R. A. Urban  
Treasurer.....J. L. Frisse  
Board of Governors: R. E. Leising, D. E. Manner, J. R. Venning

### Northern Piedmont

*Organized 1952*  
Headquarters, Greensboro, N. C.

President.....S. T. Oliver  
Vice President.....L. R. Gorrell  
Secretary.....E. D. Frazier  
Treasurer.....D. T. Waynick  
Board of Governors: W. D. Graham, Jr., Arvin Page

## Local Chapter Officers—1955 (Continued)

### Oklahoma

*Organized 1935*  
Headquarters, Oklahoma City

President..... J. R. Patten  
Vice President..... F. R. Denham  
Secretary..... A. C. Shelley  
Treasurer..... W. R. Johnson  
Board of Governors: W. J. Collins, Jr., G. E. Ervin,  
W. W. Frankfurt

### Ontario

*Organized 1922*  
Headquarters, Toronto, Ont., Canada

President..... D. L. Angus  
Vice President..... C. J. Bootes  
Secretary-Treasurer..... H. R. Roth  
Board of Governors: M. C. Bailey, R. A. Ritchie,  
Jack Thompson, Charles Torry

### Oregon

*Organized 1939*  
Headquarters, Portland

President..... Dick Blankenship  
Vice President..... W. R. Norte  
Secretary..... W. B. Hayes  
Treasurer..... G. G. Van Alst  
Board of Governors: E. E. Kelly, W. H. Oscanyan,  
J. W. Wallace, Jr.

### Ottawa Valley

*Organized 1952*  
Headquarters, Ottawa, Ont., Canada

President..... E. J. Schoenherr  
1st Vice President..... N. J. Howes  
Secretary..... D. W. Banton  
Treasurer..... C. W. Watson  
Board of Governors: G. A. Gray, J. W. Green, A.  
H. Hargreaves

### Pacific Northwest

*Organized 1928*  
Headquarters, Seattle, Wash.

President..... P. I. L. Winsor  
1st Vice President..... L. F. Christofferson  
2nd Vice President..... E. W. Triol  
Secretary..... Merrill W. McKinstry  
Treasurer..... H. M. Hendrickson  
Board of Governors: W. T. Scott and H. F.  
Warren

### Philadelphia

*Organized 1916*  
Headquarters, Philadelphia, Pa.

President..... A. M. Robertson  
1st Vice President..... C. J. Lubking  
2nd Vice President..... P. H. Yeomans  
Secretary..... R. J. Sigel  
Treasurer..... C. J. Forve  
Board of Governors: J. J. Hucker

### Pittsburgh

*Organized 1919*  
Headquarters, Pittsburgh, Pa.

President..... C. L. Benn  
Vice President..... R. M. Toucey  
Secretary..... E. H. Riesmeyer, Jr.  
Treasurer..... G. W. Cost  
Board of Governors: R. J. Doherty and B. R.  
Small

### Rocky Mountain

*Organized 1944*  
Headquarters, Denver, Colo.

President..... L. L. DeLong  
Vice President..... A. S. Widdowfield  
Secretary..... W. V. Burbank  
Treasurer..... J. V. Berger  
Board of Governors: L. V. Davis, D. D. Pearsall,  
V. F. Vallero

### Sacramento Valley

*Organized 1952*  
Headquarters, Sacramento, Calif.

President..... R. A. Sarro  
Vice President..... V. W. Thornburg  
Secretary..... M. J. Delavan  
Treasurer..... S. W. Doolittle  
Board of Governors: H. D. Brainard, E. B. Green,  
J. E. Marshall

### St. Louis

*Organized 1918*  
Headquarters, St. Louis, Mo.

President..... E. T. Clucas  
Vice President..... W. P. Norris  
Secretary..... N. J. Hubbuch  
Treasurer..... F. E. Ince  
Board of Governors: G. H. Bemart, J. J.  
Blackmore, W. D. Braden, H. J. Kipp, E. C.  
Kuntz, G. B. Pattiz

### Shreveport

*Organized 1948*  
Headquarters, Shreveport, La.

President..... R. S. Segall  
Vice President..... H. E. Scott  
Secretary..... A. L. Jones, Jr.  
Treasurer..... L. E. Kneipp  
Board of Governors: J. L. Collins, Jr., J. S. Malahy,  
Jr., J. H. Reed

### South Carolina

*Organized 1954*  
Headquarters, Columbia, S. C.

President..... R. K. Rouse  
Vice President..... J. E. McMurray  
Secretary..... B. A. Leppard  
Treasurer..... H. J. Haar, Jr.  
Board of Governors: F. A. Bailey, Jr., R. K.  
Demarest, Jr., R. F. Donovan

## Local Chapter Officers—1955 (Continued)

### South Texas

*Organized 1938*  
Headquarters, Houston

President..... F. M. Neil  
Vice President..... J. C. Lewis  
Secretary..... A. B. Ullrich, Jr.  
Treasurer..... R. S. Sandifer, C. V. Chenault\*  
Board of Governors: E. B. Appling\*\*, C. V. Chenault, J. W. Holland, D. C. McNeil

### Southern California

*Organized 1930*  
Headquarters, Los Angeles

President..... J. L. McCullough  
Vice President..... H. F. Ulovec  
Secretary..... W. J. Biggar  
Treasurer..... A. Z. Levine  
Board of Governors: J. R. Hall, I. F. Norcross, R. C. Taylor, R. W. Thomas

### Southern Piedmont

*Organized 1952*  
Headquarters, Charlotte, N. C.

President..... L. F. Lawrence  
Vice President..... J. A. Rice  
Secretary..... G. C. Garrett  
Treasurer..... G. N. Payne, Jr.  
Board of Governors: Stewart Blanton, R. S. Fullerton, C. M. Setzer, Jr.

### Southwest Texas

*Organized 1946*  
Headquarters, San Antonio

President..... F. B. Frazee  
Vice President..... Boone Crisp  
Secretary..... J. L. Rea, Jr.  
Treasurer..... C. L. Herndon  
Board of Governors: E. E. Cravens, L. H. Hornor, Jr., D. T. Kern

### Toledo

*Organized 1954*  
Headquarters, Toledo, Ohio

President..... C. F. Hoffman  
Vice President..... J. J. Heilman  
Secretary..... R. I. Fruth  
Treasurer..... R. O. Benington, Jr.  
Board of Governors: F. A. Edgington, G. H. Frost, W. D. McKittrick

### Utah

*Organized 1944*  
Headquarters, Salt Lake City

President..... R. C. Evans  
Vice President..... W. L. Stuewe  
Secretary-Treasurer..... G. E. Wright, Jr.  
Board of Governors: A. R. Curtis, A. A. Maycock, D. R. Wilde

### Virginia

*Organized 1946*  
Headquarters, Norfolk

President..... T. B. Carpenter  
Vice President..... R. L. Burton, Jr.  
Secretary..... C. G. Conaway, Jr.  
Treasurer..... D. C. Delinger  
Board of Governors: W. L. Gibson and Roy Wetherington, Jr.

### Washington, D. C.

*Organized 1935*  
Headquarters, Washington, D. C.

President..... W. C. Reamy  
Vice President..... J. H. Broome  
Secretary..... S. H. Robbins, Jr.  
Treasurer..... J. D. Crabtree  
Board of Governors: G. C. F. Asker, J. S. King, H. W. Rush

### West Texas

*Organized 1953*  
Headquarters, Lubbock

President..... R. B. Carow  
Vice President..... J. C. Wharton  
Secretary..... H. L. Mayes  
Treasurer..... D. G. Halley  
Board of Governors: G. T. Keyton, R. L. Mason, L. C. McKay

### Western Massachusetts

*Organized 1955*  
Headquarters, Springfield

President..... R. E. Cross  
Vice President..... J. E. Reed  
Secretary..... F. A. Ferraro  
Treasurer..... W. A. Boucher  
Board of Governors: L. C. Lattimer, A. M. Lovenberg, K. W. Maki, Victor Petrolatti

### Western Michigan

*Organized 1931*  
Headquarters, Grand Rapids

President..... S. M. Paganelli  
Vice President..... F. L. Murray  
Secretary..... B. J. Walter  
Treasurer..... R. J. Waalkes  
Board of Governors: F. W. Brundage, W. W. Edwards, G. W. Rynbrand

### Western New York

*Organized 1919*  
Headquarters, Buffalo

President..... C. W. Kaupp  
1st Vice President..... V. N. Harwood  
2nd Vice President..... G. E. Kuhn  
Secretary..... J. P. Guerra  
Treasurer..... M. C. Beman  
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, C. W. Stone, Q. P. Thompson

\* Filled unexpired term.

\*\* Filled unexpired term for D. C. McNeil.

## Local Chapter Officers—1955 (Concluded)

### Wisconsin

*Organized 1922*  
Headquarters, Milwaukee

President.....I. J. Rossiter  
Vice President.....H. K. Forfar  
Secretary.....R. D. Rodwell  
Treasurer.....H. W. Alyea  
Board of Governors: L. C. Plaehn, W. G.  
Schlichting, O. A. Trostel

### Special Branch

#### Switzerland

*Organized 1952*  
Headquarters, Zurich

President.....H. C. Bechtler  
Vice President.....John Frei  
Secretary.....Werner Niederer  
Treasurer.....Walter Hausler  
Auditor.....Werner Gysi

### Student Branches

#### North Carolina Staté College

*Organized 1948*  
Headquarters, Raleigh

President.....R. E. Crawford  
Vice President.....R. W. McDonald  
Secretary.....S. R. Moore  
Treasurer.....R. S. Fowler  
Reporter.....W. A. Derden

#### Texas A. & M. College

*Organized 1946*  
Headquarters, College Station

President.....H. R. Patterson, Jr.  
Vice President.....E. M. Buys, Jr.  
Secretary-Treasurer.....D. E. Verble

#### Oregon State College

*Organized 1949*  
Headquarters, Corvallis

President.....P. I. Anderson  
Secretary.....D. W. Strahan  
Treasurer.....R. L. Reiley

#### University of Detroit

*Organized 1949*  
Headquarters, Detroit, Mich.

President.....L. J. Hayes  
Vice President.....V. W. Wiktorowski  
Secretary-Treasurer.....E. F. Hoelscher, Jr.  
Reporter.....R. T. Cronin

#### Purdue University

*Organized 1949*  
Headquarters, W. Lafayette, Ind.

President.....D. B. Doucette  
Secretary.....J. W. Deabler

#### University of Toronto

*Organized 1951*  
Headquarters, Toronto, Ont., Canada



**1522**

# TRANSACTIONS

of

## AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

No. 1522

SPECIAL MEETING JANUARY 22, 1955

PHILADELPHIA, PA.

A SPECIAL meeting of the Society was called to order at 2:00 p.m. January 22, 1955 in the Burgundy Room of the Bellevue-Stratford Hotel, Philadelphia, Pa., by President L. N. Hunter. The purpose of this meeting was to give notice to the members of a proposed amendment to the By-Laws as required by Article IX of the By-Laws.

There being a quorum present as required for the conduct of business, President Hunter read the proposed amendment to **ARTICLE IV, Section 4:**

*"Section 4. Publications.* Members of all grades, except Students and dues paying members whose prorated dues amount to less than \$10.00, shall be entitled to receive the Society's TRANSACTIONS and GUIDE."

This would amend the present section **Section 4** which is as follows:

*Section 4. Dues to Constitute Subscriptions.* Of the annual dues paid by Members, Associate Members, Junior Members and Affiliates, a sum equal to the current subscription price shall be deemed to be a subscription for the JOURNAL, and all such members, except those whose annual or pro rated dues shall amount to less than Five and 50/100 dollars (\$5.50), shall be entitled to receive the Society's periodical publications. Honorary Members, Presidential Members and Life Members shall be entitled to receive the Society's publications; the Society shall subscribe for the JOURNAL in their behalf.

President Hunter then explained some of the reasons for this amendment which had been unanimously approved by the Council for submission to the membership.

The adoption of the amendment to *Article IV, Section 4* of the By-Laws means that the monthly JOURNAL of the Society, incorporated in *Heating, Piping & Air Conditioning*, will no longer go to members as part of their dues. Instead, the members will, individually and independently, subscribe for the magazine

direct from the publishers. The Society, however, will keep its dues at present levels: Members, Associate Members, Affiliates \$25.00; and Juniors \$15.00. That means, to members, an increase in dues equal to the cost of the subscription which will be a special rate of \$2.00 per year for those in the United States and Canada; \$3.00 in Mexico and the Pan American Union; and \$5.00 for foreign. These are preferential rates granted *exclusively* to Society members under a new 3-year plan which was developed by a special committee and approved by Council.

Like other organizations with rising costs, the Society must find ways of increasing its income in order to meet the cost of membership services. This arrangement will help substantially. The Society *benefits* by the retention of the money previously paid out of dues for subscriptions. It cuts out costly record keeping on subscriptions by having the publisher assume the responsibility of collections, address changes, etc. Together with these gains, we will receive, in addition, an *increased* payment from the publisher for the editorial material provided by the Society. It will be the same JOURNAL, the same *Heating, Piping & Air Conditioning*. The publisher is fully in favor of this new arrangement because he prefers to have the subscriptions from members on a voluntary, independent basis. The Council *recommends* this new plan because:

- (1) It helps solve a difficult budget problem.
- (2) It does so at the least possible cost to each member.
- (3) It does not change any of the present services to members.
- (4) It maintains a publishing connection that has been mutually satisfactory for over 25 years.

President Hunter advised the members present that they could discuss the By-Law amendment at the opening session of the 61st Annual Meeting on Monday, January 24, 1955 when it would be submitted for a vote.

There being no further discussion, the meeting was adjourned at 2:15 p.m.



**1523**

## SIXTY-FIRST ANNUAL MEETING, 1955

PHILADELPHIA, PENNSYLVANIA

THE 61st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS was held in Philadelphia, Pa., January 24-27. Attendance figures included: Members, 1389; Guests, 681; Ladies, 442. There was an international flavor to the Meeting with members present from Mexico, Australia, England, Belgium and with delegations of engineers from both Holland and West Germany.

Four technical sessions were held in the Bellevue-Stratford Hotel, meeting headquarters. Included on the program was an address by Hon. Joseph S. Clark, Jr., mayor of Philadelphia, at the welcome luncheon, as well as the entertainment feature, Delaware Valley, U.S.A. Frolic on January 24. Many of the ladies took advantage of the tour to the beautiful Longwood Gardens near Wilmington, Delaware. The 61st Annual Banquet on Wednesday night was highlighted by the address of Dr. Milton S. Eisenhower, president, The Pennsylvania State University, who spoke on The United States and Latin America. Retiring Pres. L. N. Hunter received the Past-President's Medal from Past-Pres. M. F. Blankin, while C. S. Leopold, Philadelphia, Pa., received the F. Paul Anderson Medal for 1954 in recognition of his contributions to engineering and the general health and comfort of mankind.

### FIRST SESSION, MONDAY, JANUARY 24, 9:30 A.M.

President Hunter opened the first technical session in the Clover Room of the Bellevue-Stratford Hotel and presented his report covering many of the outstanding events of the past year, mentioning the re-organization of the Society's research work, progress made on publications and the fact that a regional meeting will be held in 1955 for the first time in some years. He also reported that finances of the Society were in good condition, the number of chapters had reached a total of 59 and a regional organization plan had been under study. Membership had reached 9900, long range plans were under active consideration. He completed his report by pointing out that the change of the Society's name should be beneficial and had been carried out with the least possible change in the operations of the Society.

### TREASURER'S REPORT

The treasurer, E. R. Queer, University Park, Pa., gave an interesting summary of important figures from the Accountants' Report. His formal report included the Accountants' Report for the fiscal year ending October 31, 1954.

## Accountant's Report

### FRANK G. TUSA & CO.

CERTIFIED PUBLIC ACCOUNTANTS

37 Wall St., New York 5, N. Y.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.  
62 Worth Street  
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.—New York, N. Y. and Cleveland and its related funds for the fiscal year ended October 31, 1954 and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1954. For the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements, and the disbursements were supported by payment vouchers. The dues from members and the interest from investments were accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1954 is submitted herewith and your attention is directed to the following comments thereon:

**CASH**

The Cash on Deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the cash on hand was counted.

**MARKETABLE SECURITIES**

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safe-keeping. Securities have been included in the accompanying balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

**ACCOUNTS RECEIVABLE**

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1954 were classified and aged as follows:

MEMBERSHIP		1954
Members.....		\$ 4,324.23
Associates.....		3,993.50
Affiliates.....		4,154.25
Juniors.....		2,333.32
Students.....		12.00
<b>TOTAL.....</b>		<u><u>\$14,817.30</u></u>
<b>SUNDY DEBTORS</b>		
Unpaid charges made during 1954.....		\$ 3,926.27
Unpaid charges made during 1953.....		228.29
<b>TOTAL.....</b>		<u><u>\$ 4,154.56</u></u>

After writing off all receivables known to be uncollectible, it is our opinion that reserves for dues and accounts receivable doubtful of collection reflected in the accompanying Balance Sheet are ample to cover collection losses that may be incurred.

**RESEARCH FUND ACCOUNTS RECEIVABLE**

The balance due to the Research Fund from the Society as at October 31, 1954 represents the unpaid balance of 40% of the dues receivable from members, associates, and affiliates as of such date.

## INVENTORIES

The following inventories were verified either by physical count or direct communication.

TRANSACTIONS .....	\$7,989.59
Emblems .....	398.14
GUIDE Paper .....	915.69
Membership Roll Paper .....	200.00
<b>TOTAL .....</b>	<b><u>\$9,503.42</u></b>

A schedule of TRANSACTIONS inventories follows:

Volume	Year	Quantity	Price	Amount
1-53	1895-1947	4,151	\$ .40	\$1,660.40
54	1948	783	1.78	1,393.74
55	1949	272	2.02	549.44
56	1950	691	2.08	1,437.28
57	1951	702	2.275	1,597.05
58	1952	145	1.796	260.42
59	1953	544	2.006	1,091.26
<b>TOTAL .....</b>				<b><u>\$7,989.59</u></b>

## DEPOSITS RECEIVABLE

The deposit placed with the United Airlines in the sum of \$425.00 was verified by direct communication.

## ADVANCES

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of \$3,643.46 and \$1,331.41 respectively.

## DEFERRED CHARGES

HPAC Subscriptions .....	\$3,207.93
Insurance Premiums—Cleveland .....	440.05
Insurance Premiums—New York .....	534.07
<b>TOTAL .....</b>	<b><u>\$4,182.07</u></b>

## PERMANENT ASSETS

The Land and Buildings, Instruments, Equipment and Furniture and Fixtures are reflected on the Balance Sheet at cost of acquisition. With the exception of land and buildings all assets have been depreciated at the rate of 10% per annum. During the year fully depreciated Furniture and Fixtures in the amount of \$55.30 was written off against the reserve.

In accordance with the resolution adopted by the council at its meeting of January 23, 1949 depreciation was not provided on the Buildings for the current fiscal year.

## DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follows:

MEMBERSHIP	AMOUNT
Members .....	\$ 756.90
Associates .....	59.62
Affiliates .....	315.25
Juniors .....	461.57
Students .....	23.50
<b>TOTAL .....</b>	<b><u>\$1,616.84</u></b>

## 6 TRANSACTIONS AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

### RESERVE FOR TRANSACTIONS

Out of the original \$14,000.00 provided for TRANSACTIONS Volume 59 there remains an unexpended balance of \$13,962.43 which is reflected in the accompanying Balance Sheet.

### RESERVE FOR BUILDING MAINTENANCE

On October 31, 1952 the Council voted:

"That disbursements be made from the Building and Maintenance Reserve Fund as authorized by the Building Committee for major items of repair and maintenance, but any balance remaining in the fund beyond the expenditures for each year would go into reserve until the Building Maintenance Reserve Fund reached a total of \$10,000.00."

The fund had a \$4,000.00 balance as at the beginning of the fiscal year. An additional \$2,000.00 was provided in the budget for the current year. During the year the following major repairs were made from proceeds withdrawn from this fund:

Gutter repair, Office roof.....	\$ 723.00
New roof, Laboratory.....	898.00
Painting, one coat, office.....	641.00
TOTAL.....	<u>\$2,262.00</u>

Inasmuch as Council (October 8) authorized expenditures for laboratory repairs, the foregoing expenditures were made from the Reserve for Building Maintenance Fund, and are not reflected in the Statement of Income and Expenses of the Society. As at the end of the current fiscal year, the Building Maintenance Reserve Fund had an unexpended Balance of \$3,738.00.

### FUNDS

There is included as a part of this report an analysis of Funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1954.

### INSURANCE

The insurance coverage of the Society follows:

#### FIRE

Building (Cleveland).....		\$201,000.00
Personal Property:		
Cleveland.....	\$104,000.00	
New York.....	25,500.00	
Printers.....	12,000.00	141,500.00

#### SPRINKLER LEAKAGE

Waverly Press.....	10,000.00	
Hawes and Petit.....	2,000.00	12,000.00

#### NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE

Public Liability.....		100/300M
Property Damage.....		25M
Medical.....		1M

#### GENERAL PUBLIC LIABILITY—PREMISES

New York and Cleveland		
Liability.....		100/300M
Bodily Injury.....		25/100M

#### COMMERCIAL BLANKET BOND

All employees including the President, Treasurer and Chairman of the Finance Committee.....		20,000.00
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Respectfully submitted,  
FRANK G. TUSA & CO.  
Certified Public Accountants

## BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

New York, N. Y.—October 31, 1954

## ASSETS

GENERAL FUNDS			
SOCIETY ASSETS			
Per Schedule.....			\$77,221.12
RESEARCH ASSETS			
Per Schedule.....			61,230.32
PROPERTY FUND			
Land and Buildings.....	\$84,920.96		
Less: Reserve for Depreciation.....	3,908.07	\$81,012.89	
Laboratory and Equipment.....	70,156.49		
Less: Reserve for Depreciation.....	40,016.32	30,140.17	
Furniture and Fixtures.....	17,869.79		
Less: Reserve for Depreciation.....	6,415.64	11,454.15	
Library (Cleveland).....		300.00	
Tools (Cleveland).....		300.00	123,207.21
BUILDING FUND			
Cash on Deposit.....		851.24	
Securities			
At Cost (Market Value \$37,533.40).....	34,304.00		
Add: Accumulated Interest.....	3,191.90	37,495.90	38,347.14
BUILDING MAINTENANCE RESERVE FUND			
Cash on Deposit.....		3,600.00	
Due from Research General Fund.....		138.00	3,738.00
SOCIETY RESERVE FUND			
Cash on Deposit.....		18,444.29	
Securities at Cost (Market Value \$89,204.85).....	86,346.50		
Add: Accumulated Interest.....	2,858.35	89,204.85	107,649.14
ENDOWMENT FUND			
F. PAUL ANDERSON FUND			
Cash on Deposit.....		257.29	
Securities at Cost (Market Value \$979.00).....	1,000.00		
Add: Accrued Interest.....	12.50	1,012.50	1,269.79
RESEARCH FUND			
Cash on Deposit.....			740.03
RESEARCH RESERVE FUND			
Cash on Deposit.....			13,102.65
TOTAL ASSETS.....			<u>\$426,505.40</u>

# 8 TRANSACTIONS AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

## LIABILITIES

### GENERAL FUND

Federal Withholding Taxes.....			\$ 1,698.00
Due Research General Fund.....			3,727.70
Accrued Accounts.....			
Salaries and Commissions.....	\$12,023.73		
Professional Fees.....	1,000.00	13,023.73	
Sundry Account Payable.....			100.00
Deferred Income			
Prepaid Membership Dues			
Elected Members.....	\$1,616.84		
Candidates.....	978.25	2,595.09	
Prepaid Admission Fees.....		2,239.75	4,834.84
Reserve for Publications			
Transactions—Volume 59.....			13,962.43
Reserve for Fluctuation			
in Canadian Exchange.....		71.23	\$ 37,417.93

### RESEARCH FUND

Sundry Accounts Payable.....	116.15		
Due to Building Maintenance Reserve Fund.....	138.00	254.15	
Deferred Income			
Unexpended Earmarked Contributions.....		18,153.43	18,407.58

## NET WORTH

Society General Fund.....	39,803.19		
Research General Fund.....	42,822.74		
Property Fund.....	123,207.21		
Building Fund.....	38,347.14		
Building Maintenance Reserve Fund.....	3,738.00		
Society Reserve Fund.....	107,649.14		
F. Paul Anderson Fund.....	1,269.79		
Research Endowment Fund.....	740.03		
Research Reserve Fund.....	13,102.65	370,679.89	
TOTAL LIABILITIES AND NET WORTH.....			\$426,505.40

## COMPARATIVE STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

New York, N. Y.—For the fiscal years ended October 31, 1954 and October 31, 1953

	Fiscal Year Ended		
	October 31, 1954	October 31, 1953	Increase (Decrease)
INCOME			
Income from Dues.....	\$133,756.91	\$123,800.89	\$ 9,956.02
Admission Fees.....	8,429.50	9,690.00	( 1,260.50)
Emblems, Pins, Etc.....	819.75	1,172.00	( 352.25)

## PUBLICATIONS

JOURNAL Contracts.....	23,500.00	23,500.00	-0-
GUIDE.....	144,845.24	132,973.58	11,871.66
TRANSACTIONS.....	1,473.83	3,158.79	( 1,684.96)
Books, Reprints, Codes, Etc.....	2,026.57	3,446.80	( 1,420.23)
Income from Investments.....	1,282.89	927.09	355.80

## RESEARCH

Dues.....	69,784.26	64,381.43	5,402.83
U. S. Navy Research.....	13,340.89	34,798.45	(21,457.56)
Contributions—General.....	19,393.32	17,535.37	1,857.95
Contributions—Earmarked.....	41,079.96	29,852.00	11,227.96
Interest.....	43.87	39.26	4.61

TOTAL INCOME.....	<u>\$459,776.99</u>	<u>\$445,275.66</u>	<u>\$14,501.33</u>
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## EXPENSES

Committees and Chapters.....	\$ 24,818.44	\$ 20,995.59	\$ 3,822.85
Meetings.....	5,881.86	5,923.53	( 41.67)

## PUBLICATIONS

Members Subscriptions to <i>HPAC</i> .....	18,139.37	16,618.29	1,521.08
TRANSACTIONS.....	17,699.65	21,737.45	( 4,037.80)
Membership Roll.....	2,236.22	2,918.57	( 682.35)
Books, Reprints, Codes, Etc.....	2,557.58	2,510.91	46.67
GUIDE.....	87,217.15	83,128.64	4,088.51
Headquarters.....	143,375.21	128,687.05	14,688.16
Fund Raising.....	10,780.01	11,551.07	( 771.06)

## RESEARCH

Committee Expenses.....	4,915.63	4,210.46	705.17
Staff Salaries.....	76,849.90	75,411.51	1,438.39
Laboratory Expenses.....	20,951.41	18,340.33	2,611.08
Building Operation and Maintenance.....	9,463.39	9,454.06	9.33
Provision for Building Maintenance Reserve.....	2,000.00	2,000.00	-0-
U. S. Navy Research.....	9,851.30	24,834.26	(14,982.96)
Cooperative Research.....	33,165.59	22,025.66	11,139.93
Other Deductions.....	5,261.43	16,391.93*	(11,130.50)

TOTAL EXPENSES.....	<u>475,164.14</u>	<u>466,739.31</u>	<u>8,424.83</u>
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EXCESS OF EXPENSES OVER INCOME.....	<u>\$ 15,387.15</u>	<u>\$ 21,463.65</u>	<u>\$ 6,076.50</u>
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\* Other deductions for fiscal year ended October 31, 1953 included \$9,690.00 in Admission Fees allocated to Reserve Fund.

## REPORT OF COUNCIL

The organization meeting of the Council was held on January 28, 1954 and four regular meetings and two special meetings were held during the year.

Appointments of Committees were made in accordance with the By-Laws of the Society. A one-year vacancy on Council was filled by appointment of B. W. Farnes, Portland, Ore. and a vacancy on the Committee on Research was filled by appointment of F. K. Hick, M.D., Chicago, Ill.

A new director of research was appointed and E. R. Kaiser assumed his new duties in the middle of August.

Approval was given for having By-Laws drafted so that membership requirements would conform to suggested *ECPD* recommendations.

Council proposed a change of name for the Society to include the words, *air conditioning*, and this was adopted at a special meeting in New York on November 22. The new name, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., was filed with the Secretary of State of the State of New York was officially approved and became effective December 8, 1954.

After a publication survey authorized by Council, it was concluded that a change in method of publishing the monthly JOURNAL would be advisable so that all members could subscribe for the magazine directly to the publisher and, in order to do so, a special meeting of the Society was called for January 22 at Philadelphia to vote on a necessary By-Law change.

As required by Section 46 of the Membership Corporations' Law of the State of New York the following report is presented by the Council and filed herewith:

#### Assets

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio.....	\$ 81,012.89
Furniture and Equipment, Tools, etc., at New York and Cleveland.....	42,194.32
On deposit in New York and Cleveland banks.....	138,764.73
Securities (United States and Utility Bonds) with Bankers Trust Co., New York as custodian.....	131,523.11
Accounts Receivable.....	13,924.99
Inventory Items.....	9,503.42
Deferred Items.....	4,182.07
Miscellaneous.....	1,534.17
Total.....	\$422,639.70
1953 TOTAL.....	420,075.86
Increase.....	\$ 2,563.84

The Sources of Society Cash Receipts were: Admission Fees and Dues \$143,006.16; Publications \$171,845.64; Interest \$1,282.89; From Research \$143,642.30—Total \$459,776.99. Cash Disbursements were: Meetings, Committees and Chapters \$30,700.30; Publications \$127,849.97; Headquarters \$143,375.21; Fund Raising \$10,780.01; Research \$159,106.76—Total \$471,812.25.

The Society's Reserve Fund was increased by the addition of \$8,429.50 Admission Fees and \$2,527.62 Interest earned on Savings Bank Accounts and Securities. This is an increase of \$10,957.12 bringing the Reserve Fund to \$107,649.14.

Now in the Custodian Account of the Bankers Trust Co., New York, there are securities of the United States and Canadian Governments and a utility which cost: General Fund \$3,782.36; Building Fund \$34,304.00; Reserve Fund \$86,346.50; and F. Paul Anderson Fund \$1,000.00. The Research Reserve Fund now amounts to \$13,102.65.

#### Liabilities

The Society has the following Liabilities:

Accounts Payable.....	\$ 216.15
Federal Withholding Taxes.....	1,698.00
Accrued Accounts.....	13,023.73
Deferred Income:	
Research Projects.....	18,153.43
Prepaid Dues, Initiation Fees, etc.....	4,834.84
Reserve for	
TRANSACTIONS.....	13,962.43
Fluctuation in Canadian Exchange.....	71.23
Net Worth (Funds).....	370,679.89
TOTAL.....	\$422,639.70

At the end of the fiscal year, October 31, 1954, the Balance Sheet shows the Society's Net Worth as \$370,679.89 compared to \$381,955.09 on October 31, 1953, a net loss of \$11,275.20.

The insurance coverage of the Society is as follows: Fire for Buildings \$201,000.00; for Contents \$141,500.00; Sprinkler \$12,000.00; Automobile Comprehensive Public Liability \$100,000-300,000.00; Property Damage \$25,000.00; Medical Payments \$1,000.00; Bodily Injury \$25,000-100,000.00; Fidelity Bonds on Officers and all Employees \$20,000.00. Workmen's Compensation is carried in New York and Ohio as required by law.

The Society has contributed to the Employees Retirement Plan Trust \$2,129.85 and the participating employees contributed a similar amount deducted from salary.

Total cash in New York and Cleveland banks \$138,764.73; consisting of \$46,550.97 in General Operating Fund, \$18,444.29 in Reserve Fund, \$257.29 in F. Paul Anderson Fund, \$55,218.26 in the Research General Fund, \$13,102.65 in Research Reserve, \$740.03 in Research Endowment, \$3,600.00 in Building Maintenance Reserve Fund and \$851.24 in the Building Fund.

### *Membership*

The Society admitted the following members in the grades indicated since January 1, 1954:

Members.....	367
Associates.....	15
Affiliates.....	341
Junior Members.....	318
Students.....	106

TOTAL..... 1,147

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total is 9,920.

Respectfully submitted,

L. N. HUNTER, *President*

E. R. QUEER, *Treasurer*

President Hunter stated that the next order of business was to give attention to two proposed amendments to the By-Laws of the Society.

The first of these is an amendment to **Article VII, Section 3 (h)**, on which the members had been given notification at the Swampscott Meeting and President Hunter called for a motion on this proposed change. John Everetts, Jr., Philadelphia, Pa., moved that the proposed change in **Article VII, Section 3 (h)**, paragraph 5 of the By-Laws be adopted and this was seconded by A. W. Edwards, Cincinnati, Ohio.

President Hunter then read the proposed By-Law change which is as follows:

The Committee on Research shall recommend to the Council for appointment a Director of Research whose activities, proceedings and reports shall be subject to the direction of the Committee and the approval of the Council. The salary of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

Since there were no objections to a vote by a show of hands, the motion was put to a vote using that method and was unanimously adopted.

President Hunter then drew attention to the second By-Law change which pertains to **Article IV, Section 4** of the By-Laws, and having to do with JOURNAL subscriptions. Notification to the membership concerning this proposed change had been covered by calling a special meeting of the Society on Saturday, January 22, 1955 at 2:00 p.m. at the Bellevue-Stratford Hotel in Philadelphia.

President Hunter stated that the proposed amendment to **Article IV, Section 4** reads:

Members of all grades, except Students and dues paying members whose pro-rated dues amount to less than \$10.00, shall be entitled to receive the Society's TRANSACTIONS AND THE GUIDE.

Section Vice Pres. John W. James, Chicago, Ill., moved the adoption of this By-Law change and the motion was seconded by Mr. Everetts.

Following the discussion of the motion, a vote by secret ballot was taken. President Hunter appointed A. W. Edwards as chairman of a teller's committee to serve with R. T. Kern, Fitchburg, Mass., J. S. Locke, Minneapolis, Minn., and R. S. Dill, Washington, D. C.

When the tellers completed their count, President Hunter announced that the motion had carried by a vote of 116 for the motion and 35 against, constituting more than a 2/3 majority.

President Hunter then turned the meeting over to E. F. Snyder, Jr., Minneapolis, Minn., who presided during the presentation and discussion of three technical papers (see Program, p. 22).

Following the presentation of the papers, the session was adjourned at 12:00 noon by President Hunter.

#### SECOND TECHNICAL SESSION, TUESDAY, JANUARY 25, 9:30 A.M.

The second technical session was called to order at 9:30 a.m. by First Vice Pres. John E. Haines who introduced R. S. Dill, chairman of the Committee on Research who presented the Annual Report of that committee.

#### ANNUAL REPORT OF COMMITTEE ON RESEARCH—1954

The research program in early 1954 was handicapped by certain problems. In particular, the Laboratory Staff had been depleted by resignations and lack of suitable replacements, and no director of research had been appointed. Danger, therefore, existed that the work might be seriously impeded. Under these circumstances the Committee on Research was glad to have the advice and assistance of Council and other Society members. In particular we had the advantage of a president with an interest and much past experience in research. Pres. L. N. Hunter was a very active participant in our search for a man to conduct a broad-scale research program and, chiefly as a result of his efforts, E. R. Kaiser was appointed. I can now report that several personnel vacancies have been filled, new projects have been activated, old projects were completed with important results and the budget was increased for the current year. In general it appears that the Research Program is being and will continue to be vigorously pushed.

On August 15, Mr. Kaiser assumed his duties as director of research, with headquarters at the Research Laboratory. He is responsible for the programs at the Laboratory and the cooperating institutions, and for the research fund raising, the latter under the guidance of the Finance Committee.

C. M. Humphreys served very efficiently as acting director of research after the demise of Cyril Tasker and won the admiration of his numerous friends on the Committee and in the Society generally while carrying a double load.

#### BUDGET AND FINANCES

The research budget for the fiscal year ending October 31, 1954, was \$177,450, essentially the same level as during the previous year, and receipts equalled the budget. The expenditures were slightly under the budget because personnel was not available to undertake some of the work planned.

Plans for an expanded fund-raising campaign under Mr. Kaiser's direction are being formulated with the Finance Committee of the Society and with the advice of the Ways and Means Committee and the executive secretary. More publicity is also in prospect for the research through the combined activities of a new public relations director at the Society headquarters and the director of research.

#### LABORATORY STAFF

The Laboratory gained several new staff members in replacement of normal turnover and as additions to help activate projects. Four members of the key technical staff have been at the Laboratory for 10 years. Promotions in responsibility and title have been given to several engineers. Before the year's end a new staff organization chart was approved.

The staff will be augmented as the program grows but was balanced before the end of the year to conduct the work outlined within the funds available.

#### COMMITTEE ACTIVITIES

The Committee on Research held four meetings during the year, two at the 60th Annual Meeting in Houston, one at the Semi-Annual Meeting in Swampscott, and one at the Laboratory. In addition, the Research Executive Committee met six times. The members of the committee have also been commendably diligent in attending 31 meetings of the Technical Advisory Committees on which they also serve, and in correspondence and liaison activities with other organizations.

Twenty Technical Advisory Committees, three coordinating committees, and 14 subcommittees, organized by specific fields of activity, were manned during the year by 259 experts from industry, universities, scientific laboratories, and consulting practice. Some members served on more than one committee. A total attendance of 271 committee members and 145 visitors was recorded at the committee meetings.

The Technical Advisory Committees considered progress reports, reviewed the technical papers, and recommended projects for inclusion in the budget. The practical touch and professional contacts throughout the industry were of invaluable assistance to the researchers. The exchange of viewpoints and experience among the members at the meetings was an additional incentive to attendance.

The chairman wishes to express his appreciation to all those who gave so liberally of their experience and time to the committee work during the two years of his tenure as chairman.

In behalf of the Committee and the Technical Advisory Committees special thanks are given to Miss Ilse M. Jahn, secretary to the Committee, under whose able direction the minutes of the many meetings were excellently taken, transcribed, edited, and distributed.

#### RESEARCH POLICY AND PLANNING

The policies and procedures with respect to the functions and responsibilities of the Committee on Research, the director of research, and the Technical Advisory Commit-

tees are outlined in an Operational Guide, which has been approved by Council. The director is responsible to the Council and Society through the Committee on Research.

We are glad to report that the new director is reviewing the research papers editorially with a view to improving their understandability and value to the general membership. Authors, particularly mathematicians, often employ terms peculiar to their own fields so that many readers find it difficult if not impossible to follow their argument or even understand their conclusions.

A Long-Range Research Program has been functioning for more than a year. The research program is constantly under scrutiny and review.

#### RESEARCH PROJECTS

During the fiscal year ending October 31, 1954, there were nine active projects at the Research Laboratory in Cleveland and 14 cooperative research projects at 11 institutions over the country. The report of the director of research provides a review of these projects and the accomplishments.

#### RESEARCH PAPERS

The Research Program yielded 13 papers for presentation at meetings and for publication in the ASHAE JOURNAL SECTION, *Heating, Piping & Air Conditioning*. Six papers were presented at the 1954 Semi-Annual Meeting in Swampscott and seven were made available for the Annual Meeting in Philadelphia in January 1955.

#### REPORT OF THE DIRECTOR OF RESEARCH

The objective of the research program is the discovery of basic scientific facts and the development of useful data and relationships in heating, cooling, ventilating and air conditioning. All of the factors ranging from weather and physical environments to human response and sensations are of interest. New trends in mechanical equipment and building construction are included.

Beyond the purely scientific and applied research, better presentation of results, and greater educational and public relations efforts are part of the responsibility of the research team.

The work of the Society's Research Laboratory in Cleveland is supplemented by projects under contract with cooperating institutions.

Guidance by the Committee on Research and the Technical Advisory Committees has been invaluable to technical progress and to maintaining high morale and enthusiasm among the researchers.

#### STAFF

During 1954 the Research Laboratory obtained two additional trained staff members. Several experienced members of the staff were promoted for clearer definition of responsibility and authority.

C. M. Humphreys, formerly senior engineer, was named assistant director of research with responsibility on cooperative projects and committee activities, as well as numerous other activities. G. V. Parmelee was made senior research supervisor of heat transfer. He has achieved an international reputation in solar radiation through glass and in allied fields. L. F. Schutrum, research supervisor, is in charge of work being done in the Environment Laboratory. R. G. Huebscher, a ten-year man, was promoted to research supervisor.

#### EQUIPMENT AND FACILITIES

Two test rooms for odor studies were completed in 1954. No other major equipment was acquired. The Laboratory is well equipped with smaller instrumentation for

studies of most factors in air conditioning, but may need computing or analogue equipment for studies of periodic heat flow.

Renovation of several staff offices, repainting of the office exterior, and major roof repairs were completed.

#### AIR CLEANING

Early in 1954 the TAC on Air Cleaning adopted a new scope and objective, *to study the basic character of the contaminants present in the air and the factors influencing their removal* . . . Limited to air delivered to confined spaces (such as rooms and ventilating systems) the goal is to make available information necessary for the development of standard methods for testing and rating air-cleaning devices and to aid in the selection of air cleaners.

*Air Filtration Research Project (University of Minnesota)*: The cooperative project was established to complete the development of an improved centrifugal sedimentation apparatus for measuring particle-size distribution in samples obtained from the atmosphere, such as would enter buildings. The method was developed by K. T. Whitby and associates. The method requires representative dust samples of less than one gram.

An extensive search of the literature on air pollution has been made. A technical paper was completed for presentation. Other apparatus was also assembled to collect dust samples and to study the discoloration (blackening) effect caused by minute particulate matter.

#### AIR DISTRIBUTION

During 1954 four research projects were under way under the guidance of the TAC on Air Distribution.

*Air Flow from Annular Slots, Ceiling Plaques and Circular Diffusers (Case Institute of Technology)*: Research resulting in two papers was completed and reported.

A paper on Performance and Evaluation of Room Air Distribution has been prepared for early presentation. An extensive test plant was built for the latter studies. Flow patterns were explored for several methods of air distribution when heating with low-temperature air, as from heat-pump systems.

*Downward Projection of Heated Air (Kansas State College)*: At the 1954 Semi-Annual Meeting, a paper reported on the downward projection of air from a unit heater fitted with three different types of outlets, *viz.*: annular, deep conical, and shallow diffuser. Data on throw were correlated with conditions at the outlet, and temperature and velocity profiles over a distance of 5 ft down from the outlet were presented.

*Friction in Rectangular Branch Take-Off Fittings (Michigan State College)*: Experimental work has been completed and a paper is in preparation.

*Pitot-Tube Studies (ASHAE Research Laboratory)*: A literature search was completed to determine the possibilities and accuracies of standard and smaller Pitot tubes for use in air-flow measurements in pipes under 8-in. diam.

#### COMBUSTION

Major research with the joint financial support of the *American Gas Association*, the *Oil Heat Institute*, and *ASHAE* was started in June under the guidance of the TAC on Combustion.

*Pulsations and Resonance in Domestic Oil- and Gas-Fired Heating Equipment (Battelle Memorial Institute)*: During the first seven months Battelle conducted a survey of eight installations and measured resonance frequencies of 11 to 76 cycles per sec,—a low hum. A study of the pulsations in laboratory installations showed that a standing pressure wave existed in the furnace and gas passes, as in an organ pipe. The amplitudes were maximum in the furnace and decreased to near zero at an open check damper or back-draft diverter in the flue pipe. The frequencies could be varied by changing the position of the barometric damper on the flue pipe. Air-fuel ratio was also a factor. Motion pictures of the flames were taken for frame-by-frame study.

As the origin of the noise is at the burner and in the flame, special test furnaces are being fabricated for more detailed study of the factors. No papers have been published.

## COOLING LOAD

All of the work done during 1954 under the guidance of the TAC on Cooling Load, had as its goal the development of more accurate methods of determining instantaneous loads of cooling equipment during daily cycles of heat gains to structures.

*Air-Conditioning Thermal Circuit* (ASHAE Research Laboratory): The *thermal circuit* technique of calculating heating and cooling loads was developed at the Laboratory several years ago. The instantaneous equipment load for a single room was calculated and compared with results obtained by conventional methods which neglect thermal storage. As an extension of the problem, the heat-flow equations were rewritten to include the solar radiation directly transmitted through the glass portion of an outer wall. Further work is in progress.

The results show that the present method used in THE GUIDE overestimates the instantaneous load.

During the year the Laboratory staff became familiar with the application of the differential equation analyzer to the solution of thermal-circuit problems. The analyzer was utilized to calculate instantaneous cooling loads for a room having shaded windows and cooled by a ceiling panel in combination with an air system. Results will be presented in a future paper.

*Thermal Circuit Techniques Applied to Heating and Cooling Problems* (University of California): During the summer of 1953 the thermal performance of a one-room test house was observed under actual weather conditions for a period of several days. The house was neither heated nor cooled.

The thermal behavior of the same house, when subjected to the same diurnal cycles, was then predicted by solving the thermal circuit representing the idealized system by means of a thermal-electrical analogue. An extensive report was received and is being condensed for a technical paper.

*Periodic Heat Flow Through Roof Sections* (ASHAE Research Laboratory): Tests have been conducted to compare periodic rates of heat flow for multi-layer construction, computed by application of the theory developed by Mackey and Wright, with heat-flow rates measured under actual weather conditions. Two roof sections were tested, the heavier of which consisted of 3-in. concrete, 2-in. rigid insulation, and built-up roofing. Analysis of the data is almost completed, and a paper presenting the results will be prepared.

*Selecting Design Conditions from Summer Weather Data* (ASHAE Research Laboratory): The objective of the study was to develop a method for analyzing summer weather data for selecting design weather values of dry-bulb and wet-bulb temperatures and of solar radiation. The method was to be applicable for different types of buildings and for local weather data in different localities.

Sol-air temperature data for New Orleans were prepared previously under a cooperative agreement with Tulane University. The data were then modified to take into account the low-temperature radiant exchange with the atmosphere and the ground.

A method has been developed for analyzing local weather data to determine the design weather conditions for summer cooling. A paper has been prepared which will be presented in 1955.

## EVAPORATIVE COOLING

The TAC on Evaporative Cooling is concerned with the study of problems relating to the transfer of heat and mass in the evaporative cooling of fluids. Three subcommittees are active in studying the problems with cooling towers, evaporative condensers and evaporative air cooling.

## HEAT FLOW THROUGH GLASS

Early in 1954 work was started at the Laboratory on a study of heat flow through skylight fenestration, under the guidance of the TAC on Heat Flow Through Glass.

Tests were made on prismatic and diffusing-type glass block skylight units. Work was also done on the performance of slit-type skylight shading devices.

Experimentation on the shading of vertical sunlit glass was completed, but some analytical work still remains to be done. A paper, Convection and Radiation Gain from Windows with Slat-Type Sunshades, is being prepared.

Two other papers are in prospect. One of these will be on the combined effect of low-velocity forced convection and coincident thermal convection on the surface conductance of a smooth plate. The other will be on the calibration of pyrheliometers.

### HEAT PUMP

The TAC on the Heat Pump is interested in solar radiation as a heat source for heating structures with heat pumps. Information on means of collecting the available solar energy is desired. The solar heat absorber may be an alternate to ground coils, wells and air as a heat source.

*Solar-Energy Absorber Studies (University of Minnesota):* Work has proceeded in three areas as listed:

1. Construction and instrumentation of solar absorbers for field tests.
2. Pyrheliometer studies of solar radiation intensities on horizontal and vertical surfaces.
3. Theoretical studies of cloudless day radiation.

Construction and instrumentation of the collectors is essentially completed. During 1955 field data will be collected and compared with the results of analytical studies.

Two pyrheliometers were obtained and calibrated and readings of solar radiation intensities on horizontal and vertical surfaces will be made during 1955.

Theoretical studies of cloudless day radiation have been completed, and a paper has been presented.

### HEATING LOAD

*Study of Infiltration in a Residence Using the Tracer-Gas Technique (University of Illinois):* Some time ago it was proposed that infiltration be studied by measuring the rate of decline in concentration of a tracer gas, such as helium, introduced into a room or building. A gas analyzer was purchased for this purpose and installed in the Warm Air Research Residence No. 2, at the University of Illinois. Tests will be made during the 1954-55 heating season under a variety of weather conditions. Infiltration rates, as determined by the rate of decay of helium content, are to be correlated with the indoor-outdoor pressure differential.

### HOT WATER AND STEAM HEATING

Included under the guidance of the TAC on Hot Water and Steam Heating are four projects on noise in piping, gas in hot-water systems, and the metastable state of water.

*Noise in Piping Systems (Northwestern University):* To learn more about the cause and transmission of noise in piping, a cooperative research project was undertaken. Instrumentation has been developed to measure sounds in the water and in the air. Results of tests were reported on sound made with a 20-ft. test section of 1-in. pipe, with flow through pipe, orifices, elbows and valves. Additional testing has been done at higher temperatures and an additional pipe size.

*Metastable State of Water (Northwestern University):* The first step was to demonstrate that water exists in a metastable state. This was first shown in a glass vessel, and by gradually reducing the pressure during heating, the condition was also produced at lower magnitude in a metal vessel with a smooth inner surface.

It is too early to establish whether the metastable condition can be reproduced in larger metal vessels to be a factor in boilers, but the investigation is continuing.

*Air Entrainment in Hot-Water Heating Systems (University of Florida):* The entrainment of air or other gases in horizontal runs of piping was demonstrated in a convector system of a University dormitory. The gas was vented through automatic air vents.

The investigation is being continued in the laboratory with a complex piping system and convectors to determine whether entrainment could be used to carry the gas back to the boiler, where it could be separated from the water and put into the compression tank.

*Venting of Hot-Water Heating Systems (University of Illinois):* The objectives of the project are to investigate the factors affecting (1) the formation of gas, (2) the separa-

tion of gas from the water in a one-pipe forced-circulation hot-water heating system, and (3) the entrainment of gas by water as it moves through the system.

A test system has been constructed for the study and the calibration of flow meters, and other instrumentation is essentially completed. Glass inspection sections are incorporated to permit the observation of gas bubble formation and movement.

Tests will be made at various water temperatures, velocities and pressures under several different cycles of operation. Chemical analyses will be made of the water in the system and of the gases removed from it.

#### HUMAN CALORIMETRY

Under the guidance of the TAC on Human Calorimetry, the Laboratory constructed and tested a human calorimeter for the Naval Medical Research Center at Bethesda, Md. The equipment has been moved to and installed at the Center.

#### INDUSTRIAL ENVIRONMENT

*Design of Local Exhaust Ventilation for Hot Industrial Processes (University of Pittsburgh):* The objective of the project, under the guidance of the TAC on Industrial Environment, is to relate the volume and pattern of the convected air stream arising from a hot body with the rate of heat release, surface temperature and physical dimensions of the body. The relation between the convection flow and the requirements of exhaust ventilation for canopy and lateral hoods is also to be determined.

Progress on this difficult project was reported in 1953, but no definite results have been reported for 1954.

#### INSULATION

Under the guidance of the TAC on Insulation, the Laboratory staff collected thermal-conductivity data ( $k$  factors) on insulating materials for THE GUIDE. The TAC reviewed the data and prepared tables of recommended design values for use when calculating heat transfer through building materials and pipe insulation materials. These tables are included in THE GUIDE, 1955.

#### ODORS

A study of odor problems, including the formation, methods of measurement and control, and the physiological reactions to them is the objective of the TAC on Odors.

The project on the effect of temperature and humidity on the perception of odors was begun in July. After training a panel of staff members to observe odor intensity and quality, tests were conducted with three odorants over a range of air humidities and temperatures.

Preliminary observations showed that air temperature had a minor effect, while absolute humidity had a significant effect, the higher humidities causing a reduction in observed odor intensity.

Quantitative verification of the adaptation to odors in a few minutes by human subjects was also obtained by noting the decline in intensity of odor perception with time of exposure. Constant odor intensity was maintained in the room.

*Ventilation Requirements for Cigarette Smoke (Harvard School of Public Health):* A paper on this subject was prepared by C. P. Yaglou. It describes the effects on occupants with ventilation rates of 6 to 60 cfm per smoker.

#### PANEL HEATING AND COOLING

Several years ago a special room was constructed in the ASHAE Research Laboratory for the study of heat transfer with a panel-heated space. Numerous papers have resulted from the program in this Environment Laboratory, and the program is scheduled for completion in the spring of 1955.

Incidental to the study, measurements of angular emissivity of heat from painted surfaces, oxidized metals and roofing asphalt were made and reported.

Work will be in progress to investigate the effect of ceiling-panel cooling with lights and internal heat sources in the room, the effect of ceiling height, and the influence of cooled air introduced through a wall register and ceiling diffusers.

A paper has been drafted on design procedures for ceiling panels, and one is being written for floor panels.

The research has had the guidance of the TAC on Panel Heating and Cooling, and four subcommittees.

#### PHYSIOLOGICAL RESEARCH

*Physiological Research (University of Illinois Medical School):* The paper, Physiological Responses to Sudden Changes in Atmospheric Environment: Studies of Normal Subjects, Obese, Hyperthyroid and Hypothyroid Patients, presented in June, completed the project.

#### PLANT AND ANIMAL HUSBANDRY

The possibilities of air conditioning on plant and animal growth and product yields have hardly been touched. The TAC on Plant and Animal Husbandry is considering the scope of the subject which would be appropriate to the Society and is expected to suggest programs in the near future.

#### SENSATIONS OF COMFORT

The classical work of Houghton and others has been discussed by the TAC on Sensations of Comfort with a view toward new studies. The effect of radiation, air velocity, temperature and humidity is to be investigated at the Laboratory when the Environment Laboratory is available. Two psychrometric rooms will be provided by partitioning the present room.

#### SORPTION

*Dynamic Characteristics of a Solid Adsorbent (Pennsylvania State University):* Although the ability of certain adsorbent materials to remove moisture from air is known and utilized by engineers, very little basic information regarding the physics of sorption is available.

A method of determining and expressing the performance characteristics of a solid adsorbent was developed. The basic factors of practical sorbent processes were investigated experimentally, and the results were expressed graphically and in equations. The results, including sample calculations, were reported in a technical paper to be presented in coming months.

The TAC on Sorption guided the program.

#### SOUND AND VIBRATION CONTROL

The TAC on Sound and Vibration Control has accepted with slight modification the outline of research problems recommended by the Engineering Subcommittee of the *National Fan Manufacturers Association*.

A paper on instrumentation and measurement of sound has been prepared which outlines projects from which a selection can be made.

#### WEATHER DATA

The TAC on Weather Data has surveyed the use of weather data by hundreds of engineers. It is also in touch with the Weather Bureau and the armed services to develop a program for obtaining the weather data needed by the profession.

Mr. Dill then introduced successively the authors of the three technical papers which were presented and discussed.

The session concluded with the report of the Inspectors of Election which was presented by W. C. Reamy, chairman.

## REPORT OF INSPECTORS OF ELECTION

<i>Ballot for Officers</i>	Total
President, John E. Haines, Minneapolis, Minn.....	2142
First Vice President, John W. James, Chicago, Ill.....	2148
Second Vice President, P. B. Gordon, New York, N. Y.....	2144
Treasurer, E. R. Queer, State College, Pa.....	2147
<i>Members of Council (Three-Year Term)</i>	
B. W. Farnes, Portland, Ore.....	2145
C. B. Gamble, New Orleans, La.....	2144
W. A. Grant, Syracuse, N. Y.....	2144
D. M. Mills, Houston, Tex.....	2145
<i>Member of Council (One-Year Term)</i>	
R. T. Kern, Leominster, Mass.....	2147
<i>Committee on Research (Three-Year Term)</i>	
R. C. Chewning, Portland, Ore.....	2146
W. S. Harris, Urbana, Ill.....	2146
N. B. Hutcheon, Ottawa, Canada.....	2146
B. H. Jennings, Evanston, Ill.....	2146
R. A. Miller, Pittsburgh, Pa.....	2148
TOTAL BALLOTS RECEIVED.....	2175
TOTAL LEGAL BALLOTS.....	2148
INVALID BALLOTS.....	27

*Scattering votes:* President 6; 2nd Vice President 4; Treasurer 1; Members of Council 10; and Committee on Research 5.

Respectfully submitted,

W. C. REAMY, JR., *Chairman*

H. H. ERICKSON

C. H. SMITH

Following receipt of the report the meeting was adjourned at 12:00 noon.

## THIRD TECHNICAL SESSION, WEDNESDAY, JANUARY 26, 9:30 A.M.

The third technical session was called to order by Second Vice Pres. John W. James who introduced Treas. E. R. Queer who presided as chairman during the presentation of the technical papers.

Four technical papers were presented and discussed. Professor Queer then turned the chair over to Mr. Haines who adjourned the session at 12:00 noon.

## FOURTH TECHNICAL SESSION, THURSDAY, JANUARY 27, 9:30 A.M.

The fourth and last technical session of the meeting was called to order by Pres. L. N. Hunter at 9:30 a.m. He at once introduced Linn Helander, Manhattan, Kans., who acted as chairman of the technical session and introduced the authors of the three technical papers.

At the conclusion of the discussion of the three papers, President Hunter resumed the chair and called for the report of the Resolutions Committee. The report was presented by Reg. F. Taylor, chairman, and was adopted unanimously.

## RESOLUTIONS

WHEREAS, the first Annual Meeting of the Society under its new name, the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and the 61st Annual Meeting since its inception is now about to end, and

WHEREAS, the meeting was held in Philadelphia, a City famed for its Brotherly Love, Pigeons and a history reaching back to the time of our forefathers; a City which was the home of our Patron Saint Benjamin Franklin, and

WHEREAS, under the excellent direction of our Officers and Staff, the aims of the Society have been much advanced through the presentation of technical papers at the Technical Sessions, of value not only to the profession but also to the general public, and

WHEREAS, the Philadelphia Chapter, through its Committee on Arrangements, of which M. F. Blankin is honorary chairman, J. W. McElgin, general chairman, and John Everetts, Jr., vice chairman, has ably provided an entertaining program and has been solicitous of the welfare and comfort of its guests, and

WHEREAS, the 12th International Heating and Air-Conditioning Exposition continues its excellent record of size and quality, with each succeeding Exposition outdoing the previous one; therefore

BE IT RESOLVED that we express our sincere gratitude and thanks:

TO the Philadelphia Chapter, its officers and committees who have made this an outstanding meeting,

TO J. W. McElgin, general chairman, and John Everetts, Jr., vice chairman of the Committee on Arrangements, for their successful efforts in conducting a pleasant meeting,

TO the Honorable Joseph S. Clark, Jr., Mayor of Philadelphia and the City of Philadelphia for their kindly welcome and hospitality,

TO the hotel managements who have been thoughtful of our needs and comfort,

TO the authors of the technical papers and those who participated in the discussions,

TO the Philadelphia Chamber of Commerce and Convention Bureau for their assistance and efforts toward making this a successful meeting,

TO the daily and technical press, radio and television for the publicity given our activities, thereby informing the public of the Society's aim to improve the health and comfort of mankind,

TO the ladies of the Philadelphia Chapter for their untiring efforts in extending hospitality to the visiting ladies,

TO A. J. Nesbitt for his excellent presentation of our guests and speakers at the Wednesday night banquet and to Rev. Lewis A. Briner for his invocation,

TO Dr. Milton S. Eisenhower for his enlightening address on the United States and Latin America,

TO C. S. Leopold, for his contributions to engineering and the general health and comfort of mankind, thus furthering the aims of the Society; his accomplishments having been recognized by the award of the F. Paul Anderson Medal, and

TO Pres. L. N. Hunter, the officers and committees, and to all others who have contributed their time and efforts to insure a successful meeting.

Respectfully submitted,

The Resolutions Committee

REG. F. TAYLOR, *Chairman*, Houston, Tex.

R. S. DILL, Washington, D. C.

S. C. GALE, Vancouver, B. C.

(The installation of the officers was conducted at the Banquet on Wednesday evening.) The technical session was adjourned at 12:00 noon.

# PROGRAM-61st ANNUAL MEETING

The Bellevue-Stratford Hotel, Philadelphia, Penna.

January 24-27, 1955

Friday—January 21

- 9:30 a.m. Executive Committee (*Room 105*)
- 1:30 p.m. Finance Committee (*Room 105*)

Saturday—January 22

- 9:30 a.m. Council Meeting (*Blue Room*)
- 10:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)
- 2:00 p.m. Research Executive Committee (*Room 104*)

Sunday—January 23

- 10:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)
- 10:00 a.m. Committee on Research (*Red Room*)
- 4:00 p.m. Welcome Tea (*Clover Room*)
- 4:00 p.m. Chapters Conference Committee (*South Garden—18th Floor*)

Monday—January 24

- 9:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)
- 9:30 a.m. FIRST TECHNICAL SESSION (*Clover Room*)
  - Call to Order by Pres. L. N. Hunter
  - Report of Officers and Council
  - Amendments to By-Laws
  - E. F. Snyder, Jr., *Chairman*
  - Ventilation Requirements for Cigarette Smoke, by C. P. Yaglou, Boston Mass., presented by Prof. Yaglou.
  - Evaluation of Panel-Type Air Cleaners by Means of Atmospheric Dust, by H. A. Endres, W. T. Van Orman and R. P. Carter, Jr., Akron, Ohio, presented by Mr. Van Orman.
  - A Rapid General Purpose Centrifuge Sedimentation Method for Measurement of Size Distribution of Small Particles, Part I—Apparatus and Method, by K. T. Whitby, Minneapolis, Minn., presented by Mr. Whitby.
- 12:15 p.m. Welcome Luncheon (*Ballroom*)
  - Speaker:* Hon. Joseph S. Clark, Jr., Mayor of Philadelphia
- 2:00 p.m. Ladies Bus Trip: Historic Philadelphia or Art Museum
- 2:00 p.m. EXPOSITION—Opening of 12th International Heating and Air-Conditioning Exposition (*Commercial Museum*)
  - Exhibit will close daily at 10:00 p.m. Monday through Thursday; will close Friday at 6:00 p.m.—ASHAE Booth 68
- 2:00 p.m. TAC on Air Cleaning, A. B. Algren, *Chairman* (*Room 104*)
- 2:00 p.m. TAC on Air Distribution, W. O. Huebner, *Chairman* (*Room 105*)
- 2:00 p.m. TAC on Odors, T. H. Urdahl, *Chairman* (*Room 106*)
- 2:00 p.m. TAC on Sorption, G. L. Simpson, *Chairman* (*Room 107*)
- 2:00 p.m. Committee for Testing and Rating Low Vacuum Heating Pumps, W. M. Wallace, II, *Chairman* (*Room 108*)
- 4:00 p.m. Chapters Conference Committee (*Junior Room*)
- 4:00 p.m. Guide Committee (*Room 108*)

**Monday—January 24 (continued)**

- 6:00 p.m. Delaware Valley, U.S.A. Frolic: (*The Benjamin Franklin—Ballroom*)  
 Buffet Supper served 6:00 p.m. to 10:00 p.m.  
 Three-Act Play at 8:15 p.m., *The Man That Corrupted Hadleyburg*,  
 presented by the Palette Players of the Cheltenham Township Art  
 Center  
 Dancing—10:00 p.m.

**Tuesday—January 25**

- 9:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)  
 9:30 a.m. SECOND TECHNICAL SESSION (*Ballroom*)  
 Call to Order by First Vice Pres. John E. Haines  
 Report of Committee on Research, R. S. Dill, *Chairman*  
 R. S. Dill, *Chairman*  
 Preliminary Studies of Heat Removal by a Cooled Ceiling Panel, by  
 L. F. Schutrum, John Vouris and T. C. Min, Cleveland, Ohio, pre-  
 sented by Mr. Schutrum.  
 Measurement of Angular Emissivity, by A. Umur, G. V. Parmelee and  
 L. F. Schutrum, Cleveland, Ohio, presented by Mr. Parmelee.  
 Circuit Analysis Applied to Load Estimating, Part II—Influence of  
 Transmitted Solar Radiation, by H. B. Nottage, Santa Monica, Cal.,  
 and G. V. Parmelee, Cleveland, Ohio, presented by Mr. Parmelee.  
 Report of Inspectors of Election  
 11:00 a.m. Ladies Luncheon and Bus Trip to Longwood Gardens  
 12:00 noon EXPOSITION—12th International Heating and Air-Conditioning Exposi-  
 tion (*Commercial Museum*)  
 1:30 p.m. TAC on Industrial Environment, P. J. Marschall, *Chairman (Room 105)*  
 2:00 p.m. Committee on Research (*Room 108*)  
 2:00 p.m. Chapters Conference Committee (*Junior Room*)  
 2:00 p.m. TAC on Evaporative Cooling, (Subcommittee C) A. J. Hess, *Chairman*  
 (*Room 104*)  
 4:00 p.m. Committee on Standards for Comfort Air Conditioning, W. L. Fleisher,  
*Chairman (Room 106)*  
 7:00 p.m. Past Presidents' Dinner (*Union League Club*)  
 7:30 p.m. TAC on Cooling Load, C. O. Mackey, *Chairman (Room 104)*  
 7:30 p.m. TAC on Insulation, M. W. Keyes, *Chairman (Room 107)*  
 7:30 p.m. TAC on Sound and Vibration Control, H. A. Lockhart, *Chairman (106)*

**Wednesday—January 26**

- 9:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)  
 9:30 a.m. THIRD TECHNICAL SESSION (*Ballroom*)  
 Call to Order by Second Vice Pres. John W. James  
 Prof. E. R. Queer, *Chairman*  
 Gas is an Important Factor in the Thermal Conductivity of Most In-  
 sulating Materials, Part II, by R. M. Lander, Minneapolis, Minn.,  
 presented by Prof. R. C. Jordan, Minneapolis, Minn.  
 Selection on Outside Design Temperature for Heat Load Estimation,  
 by M. L. Ghai, Hartford, Conn., and R. Sundaram, New Delhi, India,  
 presented by Mr. Ghai.  
 A Theoretical and Experimental Study of Liquid-to-Liquid Heat Trans-  
 fer in Hot Water Heaters, by F. W. Hutchinson, Berkeley, Cal., L. J.  
 La Tart and N. W. Smith, Rome, N. Y., presented by Prof. Hutchinson.

**Wednesday—January 26 (continued)**

- Solar Radiation During Cloudless Days, by J. L. Threlkeld and R. C. Jordan, Minneapolis, Minn., presented by Prof. Threlkeld.
- 12:00 noon EXPOSITION—12th International Heating and Air-Conditioning Exposition (*Commercial Museum*)
- 12:30 p.m. Ladies Luncheon and Fashion Show (*Warwick Hotel—Ballroom*). Presented by Bonwit Teller
- 2:00 p.m. Nominating Committee (*Room 104*)
- 2:00 p.m. TAC on Heat Flow Through Glass, R. A. Miller, *Chairman (Room 106)*
- 2:00 p.m. TAC on Heat Pump, E. P. Palmatier, *Chairman (Room 107)*
- 2:00 p.m. TAC on Weather Data, John Everetts, Jr., *Chairman (Room 108)*
- 2:00 p.m. Publication Committee, Linn Helander, *Chairman (Room 105)*
- 7:00 p.m. BANQUET (*Ballroom*)  
*Toastmaster:* A. J. Nesbitt  
*Invocation:* Rev. Lewis A. Briner, Calvary Presbyterian Church, Wyncote, Pa.  
 Presentation of F. Paul Anderson Medal to C. S. Leopold by L. N. Hunter, President, ASHAE  
 50 Year Scrolls:  
 S. R. Lewis, Chicago, Ill., H. C. Meyer, Jr., New York, N. Y. and W. R. Stockwell, Michigan City, Ind.  
 Installation of Officers  
 Presentation of Past President's Emblem to L. N. Hunter by M. F. Blankin  
*Speaker:* Dr. Milton S. Eisenhower, President, The Pennsylvania State University  
*Subject:* The United States and Latin America

**Thursday—January 27**

- 9:00 a.m. REGISTRATION (*Lobby Floor—Burgundy Room*)
- 9:30 a.m. FOURTH TECHNICAL SESSION (*Ballroom*)  
 Call to Order by Pres. L. N. Hunter  
 Prof. Linn Helander, *Chairman*  
 Paths of Horizontally Projected Heated and Chilled Air Jets, by Alfred Koestel, Cleveland, Ohio, presented by Prof. Koestel.  
 Air Conditioning of Multi-Room Buildings, by R. W. Waterfill, New York, N. Y., presented by Mr. Waterfill.  
 Effects of Weather Conditions on Cooling Unit Operation in a Residence, by H. T. Gilkey, W. S. Stoecker and S. Konzo, Urbana, Ill., presented by Mr. Gilkey.  
 Report of Committee on Resolutions  
 Unfinished Business  
 New Business  
 Adjournment
- 10:00 a.m. Ladies Bus Trip to Atlantic City: Luncheon (*Haddon Hall—Wedgewood Room*), Boardwalk Sightseeing
- 12:00 noon EXPOSITION—12th International Heating and Air-Conditioning Exposition (*Commercial Museum*)
- 1:00 p.m. Organization Meeting of 1955 Council (*Pink Room*)

**Friday—January 28**

- 12:00 noon EXPOSITION—12th International Heating and Air-Conditioning Exposition (*Commercial Museum*) (Closes at 6:00 p.m.)



**1524**

## VENTILATION REQUIREMENTS FOR CIGARETTE SMOKE

By C. P. YAGLOU\*, BOSTON, MASS.

ANALYSES of tobacco and its smoke for toxic substances have been made by many investigators, notably by the Connecticut Agricultural Experiment Station<sup>1</sup>, by Baumberger<sup>2, 3</sup>, Bogen<sup>4</sup>, Wolman<sup>5</sup>, and Bradford, et al<sup>6</sup>. The principal injurious substances found in tobacco smoke are the oily alkaloids, nicotine, pyridine and its derivatives; ammonia, aldehydes, tar and resinous compounds, smoke particles, carbon monoxide and various volatile acids. No average values can be quoted for most of these substances as the composition varies with variety of tobacco, manner of smoking, and other variables, including moisture content of the tobacco and the air.

Until recently when an association was found between smoking and lung cancer, physicians considered nicotine to be the most potent poison in the smoke. It is now generally believed that all of the substances enumerated above contribute, more or less, to the toxicity of smoke.

The burning end of a cigarette was found to give off most of the toxic material likely to affect the non-smokers<sup>4</sup>, and presumably the ventilation requirements of confined spaces. Pyridine and ammonia, the two strongest local irritants in the smoke, are produced at this point. A portion of the nicotine in the tobacco is decomposed by heat into pyridine, and another portion is given off in the vapor phase at the burning point. Of the nicotine sucked through, a portion condenses in the unburned cigarette butt, another portion is absorbed on the respiratory walls, and only a relatively small quantity is exhaled.

A review of the literature cited shows a nicotine content of popular cigarette tobaccos varying from 1.0 to 3.5 percent of moisture-free tobacco weight, or from 10 to 35 mg in a standard cigarette weighing about 1 gm. The amount of nicotine in the smoke is placed at between 0.4 and 1 percent of tobacco weight (4 to 10 mg in a standard cigarette). From 15 to 35 percent of this amount appears in the smoke reaching the mouth, and about 65 percent of the nicotine entering the mouth may be retained in *puffing*, or as much as 90 percent in inhaling.<sup>2, 3</sup>

The immediate effects of exposure to high concentrations of tobacco smoke are irritation of the eyes and respiratory mucosa due to combustion products of tobacco and cigarette paper, and constriction of small blood vessels cutting down the blood supply to various parts of the body. The cardiovascular changes are ascribed to a poisonous action of nicotine on nerve endings controlling blood flow. They have been repeatedly demonstrated in the literature by a rise of systolic and dia-

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<sup>1</sup> Exponent numerals refer to References.

Presented at the 61st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1955.

stolic blood pressures, an increased heart rate, and a drop of the skin temperature of the extremities, which explains the slight chill in the hands and feet sometimes experienced by normal persons after smoking. Smoking corn-silk cigarettes, which contain no nicotine, failed to produce such vascular changes, while the intravenous ingestion of nicotine in amounts comparable to those absorbed in smoking standard cigarettes, produced analogous effects<sup>7, 8</sup>.

Acute, as well as chronic, effects of tobacco smoke are not confined to smokers alone, but may appear in non-smokers who inhale the smoke of others, especially in persons who are allergic to tobacco smoke<sup>9, 10, 11</sup>.

The ventilating engineer is interested chiefly in the short-term functional effects, giving warning of more serious effects that may follow if the early signs were not heeded. The presence of unpleasant tobacco smoke odors as perceived by non-smokers and, to a greater degree by visitors upon entering a smoking room, gives the first evidence of inadequate ventilation. The odor is not due to pyridine alone, but to many odors from ammonia, aldehydes, nicotine, tarry matter, volatile acids, and other substances in the smoke.

The appearance of irritation in the eyes, nose, or throat, with smarting and lachrymation, coughing, or sneezing among non-smokers, is a sign of harmful ventilation conditions. Although ammonia and pyridine are the strongest local irritants, many of the other substances in the smoke, including nicotine, contribute to irritation. Much more important than the local irritation of the conjunctiva, from the medical standpoint, is the effect of nicotine on the optic nerves capable of causing impairment of vision in heavy smokers<sup>10</sup>, who are affected more by the fresh smoke they inhale than by the diluted smoke in the air of a room.

#### EVALUATION OF TOBACCO SMOKE ODORS

There is no satisfactory technique for measuring objectively the strength of tobacco smoke odors as perceived by the sense of smell. Various types of *osmometers* have been designed to evaluate odor strength by determining the amount of odor-free air required to dilute an odor to the olfactory threshold. All of these instrumental aids have drawbacks. They require the use of nose catheters, or blasts of air into the nostrils, which seriously interfere with the normal perception of odors. Particles of tobacco smoke are adsorbed on interior wall surfaces of the instrument, and must be scrubbed off after each inhalation.

The possibility of developing an objective physical or chemical indicator of tobacco smoke concentration, or of its odor strength, also seems remote, because tobacco smoke is a complex of many known and unknown odorous substances whose concentration varies widely with the variety of tobacco, the age of the smoke, and the manner in which a cigarette is smoked. Recently, Gant and Shaw<sup>12</sup>, Leopold<sup>13</sup>, Kuehner<sup>14</sup> and others made some interesting exploratory studies along these lines but they have not gone far enough to ascertain the practical usefulness and limitations of their methods.

In the present study the strength of tobacco smoke odor was evaluated by the sense of smell alone, according to an arbitrary 5-point scale shown in Table 1. The same scale was used for evaluating irritation on the eyes, nose, or throat. Aside from simple tests for carbon monoxide, no chemical measurements were made of any of the other toxic substances in the air.

In using the sensory odor scale (Table 1), the observers first breathed odor-free air for at least 10 min to rid their olfactory membranes of any residual odor, before *sniffing* the smoky air. One or two sniffs produced the strongest odor perception,

after which the sensation diminished owing to olfactory fatigue. The breathing of odor-free air again, restored sensitivity. For full response the nose had to be neither too dry nor too moist, and the observers had to abstain from smoking or eating for at least one hour before testing. Persons with colds, deflected septums, or injuries to the upper respiratory or olfactory mucosa were disqualified as observers. The agreement between observers, after a little practice, was within 1 point on the odor scale. A reproducible average was obtained by using six or more qualified observers.

TABLE 1—SENSORY SCALE FOR STRENGTH OF TOBACCO-SMOKE ODOR, OR FOR DEGREE OF IRRITATION TO EYES, NOSE, AND THROAT

DEGREE, ODOR OR IRRITATION	DESCRIPTION
0	<i>Imperceptible</i> odor, or irritation.
1	Perceptible odor, or irritation, but <i>not objectionable</i> .
2	Moderate odor, or irritation, little or no objection, <i>acceptable</i> level.
3	<i>Objectionable</i> odor, or irritation; condition regarded with disfavor.
4	Strong odor, or irritation, but <i>endurable</i> .
5	Very strong odor, or irritation, <i>intolerable</i> .

Note to Subjects and Observers: Record separately strength of odor and irritation, stating also locus of irritation. Record occurrence of headache, if any, and give an overall opinion as to acceptability.

Excellent reviews of recent knowledge on the anatomy and physiology of olfaction, and on subjective and objective measurement techniques, are those by McCord and Witheridge<sup>15</sup>, and by Miner and Turk.<sup>16</sup>

#### PROCEDURE

The experiments were made in two adjoining rooms, described in conjunction with our previous work on ventilation requirements for body odors<sup>17, 18</sup>. These rooms are nearly identical, having a floor area of 155 sq ft, a ceiling height of 9 ft 3 in., a window area of 20 sq ft, and a net air space of 1410 cu ft. The windows and doors are weatherstripped, and all cracks are sealed with adhesive tape. Walls and ceilings are finished with two coats of scrubable paint. The floors are covered with smooth battleship linoleum kept unwaxed for the tests. Individual air-conditioning systems can maintain the two rooms at a comfortable temperature and humidity, regardless of weather. The downward distribution system is used, with the air entering through perforations near the ceiling and leaving at floor level. Variable speed fans (D.C. motors) allow a wide range in the quantity of air supplied which is measured by orifice meters.

In the present study, one of these rooms was used as a test room, where smoking and non-smoking subjects exposed themselves to various concentrations of tobacco smoke, while sitting on armchairs, and engaging in reading, writing or conversing. The other room served as a control room for the observers on the odor panel, who appraised the strength of odor and irritation, if any, according to the scale in Table 1, by passing from the control to the test room twice every hour, or so. The control room was kept odor-free by circulating outside air at a rate of 15 changes per hour, and by restricting the number of observers occupying the room to not more than three at a time. A small, tight-closing door equipped with a curtain trap allowed access between the two rooms.

The thermal conditions in the two rooms were approximately identical in any one test. They were adjusted according to the comfort of the subjects, and varied from 72 F to 76 F with 25 to 35 percent relative humidity in the winter, and from 75 F to 81 F with 40 to 65 percent relative humidity in the summer.

Usually nine subjects took part in each test, three non-smokers who merely exposed themselves to the smoke of others, and six smokers. Only three of the latter smoked at any one time, taking turns with the three reserves. As a rule, subjects were not allowed to smoke more than 4 cigarettes per hour, and never more than 5 in emergencies when some one of the smokers failed to come to the test. In long tests, some of the smokers had to be relieved by new subjects near the middle of tests. The subjects smoked their favorite brand of cigarettes in their own accustomed manner. The total number of cigarettes smoked per hour was approximately 24 (23.6), and the average time in consuming one cigarette down to the last  $\frac{3}{4}$  in. was about  $7\frac{1}{2}$  min.

In addition to the observers' records, all of the subjects kept records of their own, using the standard sensory scale shown in Table 1. Each test consumed from 2 to 4 hr, depending on the time required to attain a steady odor level. Altogether, 34 men participated as subjects and 15 as observers in 16 tests. Thirty of the subjects were junior or senior Harvard Medical School students; the remaining four were older men working at the School of Public Health. The observers were nearly equally students and employees.

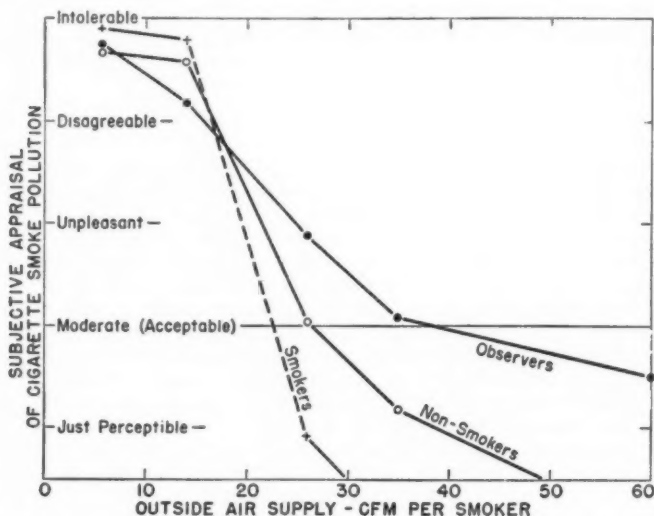


FIG. 1. SUBJECTIVE APPRAISAL OF TOBACCO SMOKE POLLUTION PRODUCED BY THREE SMOKERS IN AN EXPERIMENTAL SMOKING ROOM (10.0 x 15.5 x 9.25 FT) AT VARIOUS RATES OF AIR FLOW. NON-SMOKING OCCUPANTS OF ROOM MERELY BREATHED THE SMOKE IN THE AIR. THE OBSERVERS (VISITORS) RECORDED THEIR IMPRESSIONS IMMEDIATELY UPON ENTERING THE ROOM

## RESULTS

*Outside Air Supply in Relation to Strength of Odor and Irritation Effects:* Results of five series of tests with average outside air supplies of 6, 14, 26, 35, and 60 cfm per smoker are shown in Table 2 and Fig. 1.

An outside air supply of 35 to 40 cfm per smoker was required to remove objectionable cigarette smoke odors, as judged by observers upon entering the smoking room from odor-free air. The odors involved were those emanating from relatively

TABLE 2—SUMMARY OF VENTILATION REQUIREMENTS FOR CONTROLLING DIS-AGREEABLE EFFECTS OF TOBACCO SMOKE ON THE OLFACTORY SYSTEM, THE EYES, NOSE, AND THROAT

## EXPERIMENTAL CONDITIONS

Net floor area of smoking room	155 sq ft
Net air space of smoking room	1410 cu ft
Number of subjects smoking simultaneously	3 to 4
Total number of subjects exposed to tobacco smoke in each test	8 to 9
Duration of tests	2 to 4 hr

## OUTSIDE AIR SUPPLY 6 CFM PER SMOKER (no recirculation)

Thick bluish haze in the air.

Smarting and watering of eyes in all smokers, and most non-smokers and observers (visitors).

Irritation of nose and throat, as above, with occasional coughing.

Headache after 2 to 4 hours in all smokers and some non-smokers.

Tobacco smoke odor imperceptible, as such, to all.

Measurable increase of pulse rate and blood pressure.

Conditions strongly objectionable to all concerned.

## OUTSIDE AIR SUPPLY 14 CFM PER SMOKER (no recirculation)

Visible smoke haze in the air.

Dryness or irritation of eyes, nose, or throat of smokers, and of some non-smokers and observers.

Possible headache in few smokers and non-smokers.

Tobacco smoke odor difficult to perceive, as such.

Conditions unacceptable to all concerned.

## OUTSIDE AIR SUPPLY 26 CFM PER SMOKER (no recirculation)

Occasional trace of smoke in the air (spotty).

Possible dryness in nose or throat.

No definite effect on eyes.

Odor imperceptible to smokers, acceptable to most non-smokers, and somewhat objectionable to most of the observers.

## OUTSIDE AIR SUPPLY 35 CFM PER SMOKER (no recirculation)

Air visibly clear of tobacco smoke.

No effects on eyes, nose, or throat.

Odor imperceptible to smokers and non-smokers, and acceptable to all but a few of the observers.

## OUTSIDE AIR SUPPLY 60 CFM PER SMOKER (no recirculation)

Air visibly clear of tobacco smoke.

No effects on eyes, nose, or throat.

Odor imperceptible to all subjects, and only slightly above threshold level in the opinion of observers.

fresh tobacco smoke†. Body odors seldom could be smelled in the presence of smoke odors. Not much was gained by increasing the air flow beyond 35 cfm per smoker under the experimental conditions (Fig. 1).

Non-smoking subjects, who merely exposed themselves to the smoke of others, considered as acceptable an air supply of only 25 cfm per smoker on the basis of odor alone, which they could perceive, though less keenly than the observers, until the air supply was cut down to 14 cfm per smoker. At this reduced air flow there was distinct irritation of the eyes, nose and throat, strong enough to interfere with sense of smell. Headaches appeared when the air supply was reduced to 6 cfm per smoker, after exposures of one to three hours. The dense smoke in the air produced a feeling of depression and loss of concentration power for reading.

Although the smokers were unable to smell the smoke odor, as such, they were more susceptible to headaches and irritation of eyes, nose and throat than the non-smokers, at the lowest two air flows (Table 2).

In the light of this experience, it would seem that ventilation requirements for tobacco smoke should be based on the amount of tobacco actually consumed in a unit of time, rather than on the total number of occupants, smokers and non-smokers, as is sometimes done in practice without taking into consideration a suitable load factor. Since body odors normally cannot be smelled in the presence of tobacco smoke odor, a non-smoker actually helps in reducing the pollution, instead of contributing to it, by absorbing some of the smoke in his respiratory tract and clothing.

*Carbon Monoxide Concentration in Smoking Room:* The CO content of tobacco smoke presented no problem for the non-smoker in this study. Although concentration of this gas at the burning end of cigarettes sometimes reached values as high as 200 ppm, only a trace of less than 5 ppm could be detected in the breathing zone of the room after 4 hr of continuous smoking with ventilation rates of from 5 to 7 cfm per smoker. The maximum allowable concentration of CO is 100 ppm for exposures of 8 hr a day. A review of the literature indicates that even in heavy smokers, the accumulation of CO in their blood is not particularly significant<sup>19</sup>.

*Application and Limitations of Findings:* The results presented here are not directly applicable to practical situations, but should serve only as guides. They were obtained under more or less ideal conditions, in a relatively small room, that was kept always clean, and was supplied with air from a clean ventilating system through which no smoky air ever was recirculated during the tests. The findings do not apply to systems using recirculated air that has been treated by processes capable of altering the natural composition of tobacco smoke, such as, filtration, adsorption, spray washing, dehumidification, cooling with coils, etc.

No data are available in the literature on the allowable toxic threshold of nicotine, to ascertain whether the proposed dilution rates for controlling odors, or irritation effects are, in fact, adequate to control possible nicotine effects on non-smokers. According to the literature the most prominent functional effect of nicotine is that on the cardiovascular system. An attempt was made to correlate ventilation rates with increases of pulse rates and blood pressures observed in our experiments, but the changes were too small and too variable to justify conclusions.

It is hoped that research now in progress, or under consideration, by the Tobacco Industry Research Committee, and the American Cancer Society, will provide the ventilating engineer with the toxicological data he needs.

† The characteristics of stale tobacco smoke have been discussed in a previous report<sup>18</sup>.

## CONCLUSIONS

An outside air supply of 35 to 40 cfm per smoker was required in order to remove objectionable odors of fresh cigarette smoke, as judged by observers upon entering an experimental smoking room ( $10.0 \times 15.5 \times 9.25$  ft) from clean air. The rate of smoking was kept constant at about 24 cigarettes per hour.

Non-smoking subjects who merely inhaled the smoke in the air considered as acceptable an outside air supply of 25 cfm per smoker. The smokers themselves were incapable of perceiving the smoke odor, as such, regardless of air flow. But they were more susceptible to irritation of eyes, nose and throat, than the non-smokers or observers, when the ventilation became deficient.

Distinctly unhealthful conditions resulted when the air supply was reduced to less than 15 cfm per smoker, with headaches, smarting and watering of the eyes, irritation of the nose and throat, a feeling of depression, and a loss of concentration power for reading, among both smokers and non-smokers.

The carbon monoxide concentration was much too small to affect the non-smoker, even at the lowest air flow of 5 cfm per smoker, when the room was filled with bluish smoke.

There is no way to ascertain now whether the ventilation requirements for controlling smoke, odor, and irritation effects are in fact adequate to protect a non-smoker from possible effects of nicotine and of other poisonous smoke ingredients of unknown toxicological thresholds.

## ACKNOWLEDGMENTS

Much credit is due to E. C. Riley for his assistance in initiating this study, and to the many Harvard Medical School students and employees of the School of Public Health who served as subjects or observers.

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## DISCUSSION

LESTER AVERY, Cleveland, Ohio: I have not had the opportunity of reading the paper, and the answer may be here, Dr. Yaglou, but I raise a question at what point did you bring in air and at what point did you exhaust air?

We find in handling smoke that any effort to bring in supply air above the breathing zone is partial dilution, but never clean air.

If you bring air in from the bottom and exhaust from the top, your problem is entirely different and easily solved. I would like to know how the air was brought in.

AUTHOR'S CLOSURE (Professor Yaglou): The air was introduced through a perforated ceiling duct, and exhausted near floor level. Mr. Avery presumably feels that the main purpose of ventilating smoking rooms is to remove the smoke particles. On this presumption, an upward ventilation system should prove more efficient than a downward one by utilizing the natural tendency of fresh smoke particle to rise, and by precluding much flocculation and settling of the particles.

Unfortunately the problem is not so simple as Mr. Avery thinks. The main objective in the ventilation of ordinary smoking rooms is to remove the toxic and pungent ingredients of tobacco smoke, including nicotine, pyridine and volatile acids. All of these substances have high molecular weights and a strong tendency to sink to lower levels. They exist predominantly as vapors or gases in the air, independently of smoke particles, and cannot therefore be disposed of merely by removing the smoke particles. This can be demonstrated by passing the smoky air through an absolute filter which stops all of the smoke particles without removing the pungent gas molecules. Another convincing test may be made by forcing the air through a thick bed of activated carbon which removes most of the pungent molecules without stopping the smoke particles.

The high molecular weights of nicotine (162), pyridine (79), and acids (46-102), compared to that of air (29), make the use of a downward ventilation system imperative. Removal of these toxic substances will automatically take care of the smoke problem.

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## A RAPID GENERAL PURPOSE CENTRIFUGE SEDIMENTATION METHOD FOR MEASUREMENT OF SIZE DISTRIBUTION OF SMALL PARTICLES

### Part I-Apparatus and Method

By K. T. WHITBY\*, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn.

**T**HE RAPID increase in industrial development and concentration of urban dwellers has placed increased emphasis on the need for more effective air cleaning.

The air filtration problem generally resolves itself into a need for knowledge of contaminants present in atmospheric air. These contaminants can be broadly classified as particulate matter, gaseous matter, air-borne bacteria, viruses and odors.

The AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., through its Technical Advisory Committee on Air Cleaning† fully realizes the magnitude of this air pollution problem. The Society is, however, presently confining its activities to a study of particulate matter which exists in air that ventilates spaces used for human occupancy.

This first paper presents a new method of particle-size measurement which has been found to be of practical value for measurement in the 0.05 to 100 micron size range. Part I describes the apparatus and basic method. Part II will describe detailed procedures applicable to a variety of applications.

Problems of particle-size measurement are almost as varied as industry itself. For example, size measurement is important in the production of such substances as flour, paint pigments, powdered sugar and insecticides. In recent years the air pollution problem has received increasing attention, and a need exists for rapid yet comprehensive methods for evaluating atmospheric dust samples.

These and similar needs in other fields of particle technology have generated interest in size analyses because of the fundamental importance of size distribution data.

\* Research Associate, Mechanical Engineering Department, University of Minnesota

† Personnel: A. B. Algren, Chairman; R. B. Crepps, L. L. Dollinger, Jr., O. C. Eliason, B. G. Evans, R. S. Farr, C. D. Graham, A. J. Hess, A. B. Hubbard, G. F. Landgraf, K. W. MacKenzie, Arthur Nutting, H. E. Robinson, S. L. Root, Jr., C. B. Rowe, W. T. Van Orman, R. P. Warren, W. B. Watterson.

Presented at the 61st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1955.

The literature in size analysis is voluminous and scattered widely in various scientific journals. Lately this situation has been improved by a number of comprehensive reviews covering all methods of size analysis. Several good ones are given by Schweyer and Work<sup>1</sup>, Heywood<sup>2</sup>, Stairmand<sup>3</sup>, and Rose<sup>4</sup>. Two recent reviews which also have fairly extensive bibliographies are Herdan<sup>5</sup> and Work and Whitby<sup>6</sup>.

Study of the literature shows that only the microscopic and sedimentation methods have been used widely for the subsieve range. Microscopic analysis by electron beam or light can be used from 0.01 micron up but the procedures are tedious and inaccurate except for narrow size ranges. For this reason, sedimentation analysis has been the most widely used. All sedimentation or elutriation methods use the relationship between particle size and settling velocity derived

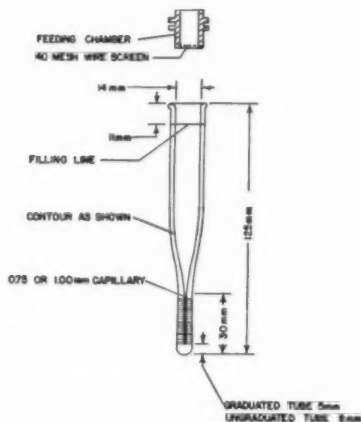


FIG. 1. CENTRIFUGE TUBE AND FEEDING CHAMBER DESIGN

from Stokes' law. Hawksley<sup>7</sup> has given an excellent discussion of the basic principles of sedimentation analysis.

Stokes' law states that the settling velocity of a particle in a fluid is proportional to the square of the particle diameter. For this reason the settling velocity of particles below 5 microns in size becomes rather slow. For example, a 1 micron particle of density 2.5 would require about 24 hr to settle 10 cm in water. Thus, for practical reasons gravity sedimentation is usually limited to 1 micron and larger. By using a centrifuge to increase the force on the particles, the sedimentation analysis may be extended to the finer size ranges. This is the approach used in the development of the method described in this paper.

Basically, the method combines gravity and centrifuge sedimentation to take advantage of the desirable characteristics of each. This approach is not new. For example, Norton and Speil<sup>8</sup> constructed a centrifuge large enough so that standard hydrometers and jars could be used.

<sup>1</sup> Exponent numerals refer to References.

Most centrifuge methods start with an initially homogeneous suspension in the tube. Robison and Martin<sup>9</sup> and Kamack<sup>10</sup> describe recent improvements in this approach. The one serious drawback in the use of a homogeneous suspension in a centrifuge is the complication involved in the calculation of results. The fact that various size particles are settling out at any given time, plus the fact that the forces on the particle vary with the radius in the centrifuge, precludes any exact solution to the differential equations that describe the rate of sedimentation. This difficulty can be eliminated by making the radius of the centrifuge large compared to the sedimentation height, or by starting all the particles off in a layer on top of the sedimentation liquid. The latter approach, first used by Marshall<sup>11</sup>, is the one used in the method described in this paper.

Several different methods have been used for sensing the rate of sedimentation. Kamack<sup>10</sup> describes an ingenious system of withdrawal pipettes which can be operated while the centrifuge is in motion. Svedberg<sup>12</sup> made extensive use of optical methods in his ultracentrifuges. Norton and Speil<sup>8</sup> used a hydrometer,

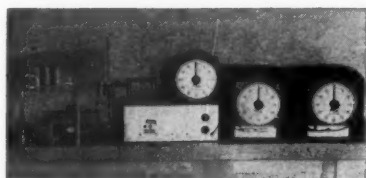


FIG. 2. COMPLETE CENTRIFUGE SIZE ANALYSIS EQUIPMENT. (L. TO R.) PROJECTION READING APPARATUS, TUBES IN RACK, MICRO STIRRER, 300 RPM CENTRIFUGE, 600-1200 RPM AND 1800 RPM CENTRIFUGE

and Berg<sup>13</sup>, a specific gravity diver. However, all of these either require elaborate apparatus or careful technique. For these and other reasons a simple capillary at the bottom of the centrifuge tube has been used to measure sediment height.

#### PRINCIPLE OF OPERATION

Basic to the method is the special centrifuge tube and feeding chamber illustrated in Fig. 1. At the beginning of a size analysis the clean tube is filled to the line near the top of the tube with a suitable sedimentation liquid. The tube is then placed in a holding device, which may be simply a laboratory clamp on a ring stand, or may be a more elaborate optical observing device as illustrated in Fig. 2.

Next a suspension of particles is made up in a liquid that is miscible with the sedimentation liquid and has a slightly lower density and a slightly higher viscosity. The particles must be practically insoluble in either liquid. An aliquot of this suspension is placed in the feeding chamber and the chamber placed in the tube and released in such a way so as to leave a sharp layer of suspension on top of the sedimentation liquid.

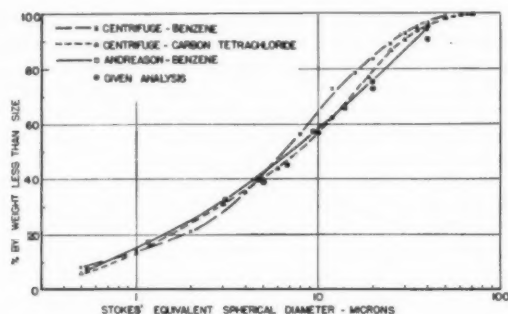


FIG. 3. PARTICLE SIZE ANALYSES OF A.C. FINE TEST DUST

Then, at times calculated from Stokes' law for the desired sizes, the height of the sediment in the capillary is read. Sedimentation is allowed to proceed under gravity for 4 to 10 min depending upon the time schedule that has previously been worked out. Then the tube is transferred to the first and lowest speed centrifuge and run for a precalculated time. The tube is removed to read the sediment height and then is replaced in the centrifuge for the next time interval. This is continued until there is no change in sediment height or until the last centrifuge time in the schedule has been run. During the centrifuge runs the speed is increased as the settling velocity of the particle decreases.

By proper choice of liquids, centrifuge speeds, and reading times, it is possible to determine size distribution at the rate of 2 per hr on such materials as the A.C.

TABLE 1—SIZE ANALYSIS DATA FOR A.C. FINE TEST DUST ANALYZED BENZENE

PARTICLE SIZE MICRONS	READING TIME MIN SEC	CENTRIFUGE R.P.M.	OBSERVED HEIGHT ON SCREEN	PERCENT GREATER THAN SIZE	PERCENT LESS THAN SIZE
60	:18		0	0	100
50	:26		0.2	0.8	99.2
40	:40		0.9	3.5	96.5
35	:53		1.3	5.0	95.0
30	1:12		2.0	7.7	92.3
25	1:43		3.0	11.5	88.5
20	2:41		4.5	17.2	82.8
16	4:12		5.9	22.6	77.4
12	:18	300	8.3	31.8	68.2
8	:24	600	11.3	43.4	56.6
4	:38	1200	16.7	64.0	36.0
2	1:03	1800	20.4	78.2	21.8
1	3:22	1800	22.4	85.8	14.2
0.5	12:34	1800	23.8	91.4	8.6
±0	15:00	4000	26.1	100.0	0

Note: Room temperature = 80 F; Sedimentation liquid = Benzene; Feeding liquid = 40 percent Benzene + 60 percent Naphtha; Dispersing agent = 0.1 percent Twitchell 8240  $\rho = 2.52$  gm per cubic centimeter;  $\rho_s = 0.87$ ;  $\eta_0 = 0.00582$  poise;  $K = 6.45 \times 10^4$ . Dispersed 2 min in a blender.

fine test dust, illustrated in Fig. 3 and Table 1. If this size analysis had been run by straight gravity sedimentation, it would have required approximately 54 hr as compared to about 25 min by centrifugal analysis.

Application of the method has been broad. Equipment and procedures are the result of the combined experience of six industrial firms and the University of Minnesota on over 50 materials ranging from paint pigments to D.D.T. insecticide.

Though extensive, the equipment required is simple enough so that some version of the method is within the reach of every laboratory. Fig. 2 illustrates the complete apparatus used in making a size analysis. The method is very flexible. Size analyses can be made on samples as small as 2 mg. One or 20 points on the distribution curve may be determined and the method is especially suitable for obtaining accurate analyses of samples containing a few percent of large particles.

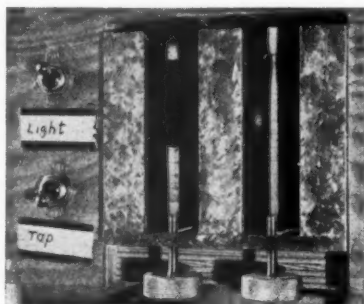


FIG. 4. PROJECTED IMAGE OF TUBE (LEFT) WITH SEDIMENT (RIGHT) EMPTY TUBE. (NOTE IMAGE IS INVERTED)

Calculation of results for routine analyses is simple, requiring only a few calculations on a slide rule. Details of the calculations involved and preparation of schedules will be described later. Selection of liquids, dispersing agents and dispersion methods are discussed in Part II (see page 449).

#### APPARATUS

*Centrifuge Tubes:* The tube design that has been developed is illustrated in Fig. 1. The graduated tubes, which can be read without any optical accessories, are necessary for the analysis of certain materials. Ungraduated tubes are satisfactory for most materials. Paint pigments and other submicron materials form an optically opaque suspension in the capillary above the sediment. Though it is possible to see the line of demarcation with the unaided eye, it is often impossible when using the projection system.

The ungraduated tubes used with a suitable projection system are desirable for most size analysis work. Magnification of the capillary image six times, as in Fig. 4, permits easier, more accurate reading of sediment height. The magnified image also makes it easy to see if dispersion is satisfactory and to see the actual shape and number of the few largest particles.

Tube size is such as to fit the standard 15 ml tube shields and heads available on most small centrifuges. Other tube dimensions have been worked out from experiment and from consideration of what it is practical to make from glass. It has been found that the two parts of the tube most difficult to make and also the most important are the transitions from the cylindrical portion to the capillary and the flat bottom of the capillary.

In addition to the tube design shown in Fig. 1, a special open-ended design having a bore of 0.17 mm is under development for use with impinger dust samples and other applications where the concentration of dust in the feeding liquid is very low.

*Centrifuges:* Early attempts to use ordinary laboratory centrifuges showed that

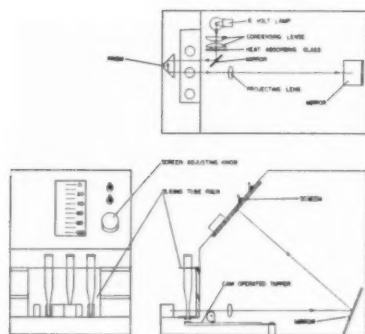


FIG. 5. SCHEMATIC DIAGRAM OF IMPROVED PROJECTION UNIT

while they can be used, they have many serious drawbacks. The major disadvantage is lack of speed stability. Acceleration rates are too low at the low speeds and too high at the high speeds. Also it is very difficult to control these rates without major reconstruction of the centrifuges.

Some of the smaller commercial centrifuges can be used if an accurate stroboscope is available and if they are operated with covers open in order to keep the tubes at room temperature and to increase aerodynamic resistance.

Considerable experimentation has established the following centrifuge characteristics as necessary:

1. Speed must be constant within 1 percent of the values used in calculating the tables.
2. Maximum rate of acceleration during starting and stopping must be less than 5 rad per (sec) (sec).
3. Speed vs time curves during starting and stopping must be known and constant enough so that starting and stopping correction will not vary more than  $\pm 0.5$  sec.
4. It is desirable to have a 300-600 rpm range for the slowest centrifuge.
5. For ease of operation it is desirable that no adjustment of rheostats or other speed control equipment be necessary during the starting and stopping period of the

centrifuge. This is to permit the centrifuge to be started and stopped by an automatic timer of suitable accuracy.

The foregoing requirements dictated the design of the special centrifuges illustrated in Fig. 2. These centrifuges are powered by multispeed hysteresis type synchronous motors. Three centrifuges have been developed so far. A 300-600 rpm unit is powered by a 600-1200 rpm motor with a 2 to 1 gear reduction between motor and head. A 600-1200 rpm unit is direct driven as is an 1800 rpm unit. Starting and stopping characteristics are controlled by a combination of an inertia disk on the motor shaft and a variable resistor in series with one winding of the motor. For some locations a wave form correcting-type voltage regulator is necessary to obtain constant starting characteristics.

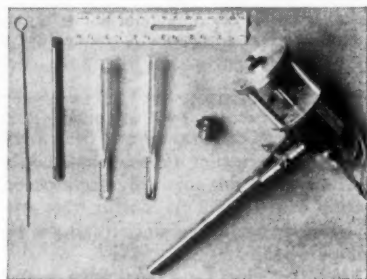


FIG. 6. MISCELLANEOUS EQUIPMENT (L. TO R.) CLEANING WIRE, POWDER SCOOP, GRADUATED TUBE, UNGRADUATED TUBE, FEEDING CHAMBER, AND MICRO-STIRRER

For convenience a 1-sec to 1-hr timer is built into each centrifuge. All of the work to date has been done with centrifuges equipped with small commercial heads. These two-place heads were used because they were the smallest standard heads available. Use of other heads with slightly different trunnion radii would have no effect other than to change the radii used in the calculation of time schedules.

*Tube-Reading Equipment:* It was found desirable to tap the tube gently on the tip along the axis of the tube during the gravity portion of the run. Therefore, for both the graduated tubes and the projection tubes, the tube holder is provided with a cam operated tapper which strikes the tube a light blow on the tip 40 times per minute.

For the ungraduated tubes a special projection system is used to read the tubes. Fig. 4 is a close-up of the screen and illustrates the type of scale and image obtained. Fig. 5 illustrates in schematic form an improved version of the optical unit which is now under construction. The tube should be cut in the open so that sedimentation can be observed in the whole tube and so that the lamp heat will not cause convection currents. Space for more than one tube is provided in a sliding rack

so that more than one analysis can be run at the same time. This is also the reason for the double optical system of the instrument illustrated in Figs. 2 and 4.

*Miscellaneous Equipment:* Several small but essential items go to complete the apparatus as follows:

1. *Feeding chamber*—as illustrated in Fig. 1—is essential for obtaining a uniform layer of suspension on the surface of the sedimentation tube.

2. Fig. 6 illustrates a *cleaning wire* and *powder scoop*. Closed end tubes are cleaned by pumping this wire rapidly in and out of capillary with the tube full of liquid. The *powder scoop* is used for measuring out a given volume of powder when the dispersion is made directly in the feeding chamber. This is particularly convenient for routine analyses on materials reasonably easy to disperse.

3. Fig. 6 also illustrates a *special micro-stirrer* that can be used to stir the suspension in the feeding chamber. This makes it unnecessary to make up a separate suspension in a blender or other stirring device. Using a motor that will develop at least 10,000 rpm, this micro-stirrer will disperse all but the most difficult materials.

4. *High speed commercial centrifuge*—A commercial centrifuge that will develop approximately 2,000 g's is useful for obtaining end points on very fine materials such as paint pigments. A small medical centrifuge is satisfactory for this purpose.

#### CALCULATION OF TABLES

Because this method combines both gravity and centrifugal sedimentation, it is most convenient to calculate the reading times completely before a run is begun. Though there are several ways of doing this, only one procedure will be given here. Developed in collaboration with Dr. M. A. Knight, this procedure is particularly adapted to industrial laboratory operations since it reduces the decisions that must be made during the computation.

*Gravity Portion of Tables:* The time for a particle to settle a given distance in a fluid under the influence of gravity can be calculated from the usual form of Stokes' law.

$$t_g = (18 \times 10^8 \eta_0 h) / [(\rho - \rho_0) g d^2] \quad . . . . . (1)$$

where

$t_g$  = time in seconds for a particle to settle a distance  $h$  under the influence of gravity.

$d$  = particle diameter, microns.

$h$  = settling height, centimeters (10 cm for all tubes).

$\rho$  = true density of particles, grams per cubic centimeter.

$\rho_0$  = density of sedimentation liquid, grams per cubic centimeter.

$\eta_0$  = absolute viscosity of sedimentation liquid, poise.

$g$  = gravitational constant.

if we let

$$K = (18 \times 10^8 \eta_0 h) / [(\rho - \rho_0) g] \quad . . . . . (2)$$

then Equation 1 becomes:

$$t_g = K/d^2 \quad . . . . . (3)$$

Since  $K$  will be a constant for a given material in a given sedimentation liquid, the gravity reading times may be calculated by multiplying the calculated  $K$  value by  $1/d^2$  values pre-calculated in tabular form. Table 2 illustrates a few  $1/d^2$  values used in calculating the time schedule of Table 3.

TABLE 2—VALUES OF  $1/d^2$  USED IN TABLE 3

$d$ -MICRONS	$1/d^2$
70	$2.04 \times 10^{-4}$
60	$2.78 \times 10^{-4}$
50	$4.00 \times 10^{-4}$
40	$6.25 \times 10^{-4}$

TABLE 3—CALCULATION OF READING SCHEDULE OF  $K = 6.45 \times 10^4$ 

RPM	VALUES OF $K_s$	VALUES OF $J_s$
300	$0.0992 K = 0.640 \times 10^4$	$100 1/K = 0.00155$
600	$0.0248 K = 0.160 \times 10^4$	$525 1/K = 0.00814$
1200	$0.0062 K = 0.040 \times 10^4$	$2900 1/K = 0.0450$
1800	$0.00276 K = 0.0178 \times 10^4$	$14,500 1/K = 0.225$

$d_e$ , IN MICRONS	$t_g$ , IN SECONDS	$d_e$ , IN MICRONS	$t_g$ , IN SECONDS	$t_g = K/d_e^2$
70	13.2	30	71.7	
60	17.9	25	103.3	
50	25.8	20	161.0	
40	40.3	16	252.0	
35	52.6			

$d_e$	$t_0^*$	$C_s$	$t_{01}$	RPM	$Q$
12	17.7	0	17.7	300	0.00277
8	18.0	+6.0	24.0	600	0.0130
4	25.4	+12.4	38.0	1200	0.0764
2	46.7	+16.0	63.0	1800	0.337
1	186.0	+16.0	202.0	1800	1.383
0.5	738.0	+16.0	754.0	1800	5.576

RPM	$t_{cn} = (Q_n - Q_{n-1})K_s$ , SECONDS
300	$0.00277 \times 0.640 \times 10^4 = 17.7$
600	$(0.0130 - 0.00277) 0.160 \times 10^4 = 18.0$
1200	$(0.0764 - 0.0130) 0.040 \times 10^4 = 25.4$
1800	$(0.337 - 0.0764) 0.0178 \times 10^4 = 46.7$
1800	$(1.383 - 0.337) 0.0178 \times 10^4 = 186.0$
1800	$(5.576 - 1.383) 0.0178 \times 10^4 = 738.0$

*Centrifuge Portion of Table:* Centrifuge settling times can be calculated by a form of Stokes' law for a centrifugal field that is derived as follows:

Let  $r$  represent the radius of rotation of a particle of mass  $m$  rotating about an axis with angular velocity  $\omega$ . At equilibrium, two forces will be acting on the particle; a frictional force determined by Stokes' law and a centrifugal force.

Equating these we obtain:

$$3 \pi \eta \omega d (dr/dt) \times 10^8 = m \omega^2 r.$$

In a liquid the effective mass of a spherical particle will be:

$$m = (\pi/6) d^3 (\rho - \rho_0)$$

Therefore:

$$dr/dt = [(\rho - \rho_0) \omega^2 d^2 r] / (18 \times 10^8 \eta_0)$$

Integrating between the starting radius  $r_1$ , and the final radius  $r_2$  we obtain:

$$t_c = t_2 - t_1 = \{ (18 \times 10^8 \eta_0) / [(\rho - \rho_0) \omega^2 d^2] \} \log_e (r_2/r_1) \quad . . . \quad (4)$$

where

$t_c$  = the centrifuge running time necessary to centrifuge a particle of size  $d$  from  $r_1$  to  $r_2$ .

In both Equations 3 and 4, the slightly different density and viscosity of the feeding and sedimentation liquid have been neglected. This approximation is valid because the thickness of the feeding layer is small and usually the density and viscosity differences are small.

The  $K$ -values may also be defined from Equation 4 if the centrifuge speeds are selected. If we let  $K_g$  represent the  $K$ -value associated with a given centrifuge speed  $s$ , then:

$$K_g = (18 \times 10^8 \eta_0) / [(\rho - \rho_0) \omega_s^2] \quad . . . . . \quad (5)$$

Equation 4 becomes:

$$t_c = (K_g/d^2) \log_e (r_2/r_1) \quad . . . . . \quad (6)$$

By equating Equations 2 and 5,  $K_g$  may be expressed in terms of  $K$  as shown in Table 3.

In Equation 6,  $r_2$  is a constant for a given tube and head design, but  $r_1$  will depend not only on the radius of the top of the tube but on how far the particle has already settled under the influence of gravity.

Therefore

$$r_1 = r_0 + 10 (t_g/t_{cg}) \quad . . . . . \quad (7)$$

where

$t_g$  = time for last gravity particle size  $d_g$  to settle 10 cm under gravity.  
 $t_{cg}$  = time for first centrifuge particle size  $d_c$  to settle 10 cm under gravity.  
 $r_0$  = starting radius of all particles in the feeding layer at  $t = 0$ .

If Equation 3 is substituted into Equation 7

$$r_1 = r_0 + 10 d_c^2/d_g^2 \quad . . . . . \quad (8)$$

We see that  $r_1$  is a function only of the last gravity size and of the centrifuge size under consideration. This makes it convenient to set up a simple calculation scheme for the centrifuge portion of the run in which use is made of a calculation constant tabulated as a function of  $d_g$  and  $d_c$ .

In Equation 6, define  $Q$  by:

$$Q = (1/d_c^2) \log_e (r_2/r_1) \quad . . . . . \quad (9)$$

then

$$t_c = K_g Q \quad . . . . . \quad (10)$$

Substitution of Equation 8 into Equation 9 yields:

$$Q = (1/d_0^2) \log_e \{r_2/[r_0 + (10d_0^2/d_2^2)]\} \quad . . . . . (11)$$

Equation 11 may then be used to calculate a table of  $Q$  values such as in Table 4, for apparatus with given  $r_0$  and  $r_2$  values.

TABLE 4—PORTION OF A TABLE OF  $Q$  VALUES USED TO CALCULATE CENTRIFUGE RUNNING TIMES<sup>a</sup>

$d_0$ MICRONS	$d_2$ MICRONS			
	16	15	14	13
14	0.00099	0.0052		
13	0.00175	0.00123	0.000649	
12	0.00277	0.00219	0.00159	0.000842
11	0.00417	0.00353	0.00281	0.00199
10	0.00613	0.00542	0.00459	0.00367
9	0.00892	0.00811	0.00719	0.00614
8	0.0130	0.0121	0.0110	0.00984
7	0.0191	0.0181	0.0169	0.0156
6	0.0288	0.0278	0.0264	0.0249
5	0.0454	0.0442	0.0427	0.0410
4	0.0764	0.0749	0.0733	0.0713
3	0.144	0.142	0.140	0.138
2	0.337	0.336	0.334	0.331
1	1.383	1.381	1.379	1.376
0.5	5.576	5.576	5.576	5.576

<sup>a</sup>  $Q$  values calculated for  $r_0 = 3.30$  cm and  $r_2 = 13.30$  cm.

Before illustrating how a table of  $Q$ 's is used to calculate a complete reading schedule, it is convenient to define one other constant. When preparing a reading schedule such as in Table 3, the question arises as to just what particle-size intervals and centrifuge speeds are desirable.

There is not too much of a problem during the gravity portion of the run because practically any size intervals may be used. However, during the centrifuge portion, a number of factors must be considered. Due to computational difficulties, it is impractical to have a centrifuge running time less than the starting time of the centrifuge at that speed. In Table 3 for example, the first centrifuge time at 12 microns is 17.8 sec. For this centrifuge the minimum useful timer setting is 10 sec. If the centrifuge time for a size of say 14 microns was to be calculated, it would be found that the centrifuge time would be only 6.3 sec, which is below the useful time and therefore impractical. The same problem exists when changing to higher speeds.

To aid in selecting the best combination of sizes and speeds, it is convenient to define the minimum useful increment in  $Q$  that can be used for each centrifuge speed.

In Equation 10,  $t_0$  is the centrifuge running time assuming that the centrifuge runs at the given speed for the whole interval. Actually a starting and stopping

correction is necessary because in most cases it is inconvenient to balance the starting time against the stopping time. Later it will be shown how this correction is obtained but for the present it is sufficient to say that it is a constant which is added algebraically to  $t_e$  in order to get the timer setting  $t_{et}$ .

Therefore,

$$t_{et} = t_e + C_s = K_s Q + C_s \quad \dots \quad (12)$$

where

$C_s$  = the starting stopping correction at speed  $S$ .

$t_{et}$  = the centrifuge timer setting necessary.

The minimum useful increment in  $Q$  may now be calculated by setting  $t_{et}$  equal to the starting time of the centrifuge.

$$t_{et} = t_{\text{starting}} = K_s J_s + C_s$$

or

$$J_s = (t_{\text{starting}} - C_s)/K_s \quad \dots \quad (13)$$

where

$J_s$  is the minimum useful increment in  $Q$  at the speed  $s$ . As an example, for the 600 rpm centrifuge  $C_s = +6$  sec, and  $t_{\text{starting}} = 19$  sec,

therefore

$$J_{600} = (19 - 6/K_{600}) = (13/0.0248K) = (525/K)$$

Table 3 illustrates how these  $Q$  and  $J$  values are used to calculate a complete schedule. A convenient step-by-step procedure is outlined.

1. Using Equation 2, calculate  $K$  for the given conditions of density and viscosity.

2. Calculate  $1/K$ ,  $K_s$  and  $J_s$  values as at the top of Table 3.

3. Calculate the gravity portion of Table 3 from Equation 3 and a suitable table of  $1/d^2$  values. It is best to terminate the gravity portion of the run between 4 and 10 min. The minimum gravity time permissible will depend on the centrifuge equipment available and the size interval from the last gravity to the first centrifuge size.

4. Select the first centrifuge size so that  $Q_1 > J_{s1}$ . In the example  $J_{300} = 0.00155$  and therefore for  $d_g = 16$  microns it is seen from Table 4 that  $d_c = 13$  microns is the maximum particle size that could be selected for the first centrifuge size. In this case 12 microns were selected because the interval from 12 to 16 microns was small enough. Calculate  $t_e$  from Equation 10, using a  $Q$  value corresponding to  $d_g = 16$  and  $d_c = 12$  and  $K_{300}$ . Add the appropriate start stop correction to get the time setting  $t_{e1}$ .

$$t_{e1} = Q_1 K_{s1} + C_{s1}$$

5. Select the next centrifuge size so that  $Q_2 > Q_1 + J_{s2}$ . The problem now arises as to whether to use the centrifuge speed used for  $d_{c1}$  or go to a higher speed. In this case  $Q_1 + J_{300} = 0.00432$  and  $Q_1 + J_{600} = 0.0109$ . From Table 4 it is seen that the maximum  $d_{c2}$  that could be used with 300 rpm centrifuge is 10 microns and with the 600 rpm centrifuge, 8 microns. Since the interval from 12 to 8 microns is satisfactory, we choose the 600 rpm centrifuge in order to save time. Then for the example:

$$\begin{aligned} t_{e1} &= Q_2 K_{600} - Q_1 K_{600} + C_{600} \\ &= 0.0130 \times 0.160 \times 10^4 - 0.00277 \times 0.160 \times 10^4 + 6.0 = 24 \text{ sec.} \end{aligned}$$

6. Subsequent centrifuge size intervals and centrifuge times are calculated using the same procedure. Because the starting and stopping of the centrifuges and the handling

of the tube causes a small amount of mixing in the tube, centrifuge reading sizes should be limited to a maximum of seven. Actual experience has indicated that five is usually sufficient.

Where many size analyses must be run on a variety of materials it is convenient to make up tables of appropriate reading schedules calculated for increments of  $K$ . Tables calculated for successive increments having a ratio of 1.025 over a range from  $5 \times 10^4$  to  $25 \times 10^4$  will cover the most used range.

#### DETERMINATION OF STARTING AND STOPPING CORRECTION

Due to the short runs that are made with the centrifuge, the starting and stopping correction becomes an appreciable fraction of the running time. This makes a precise determination of the correction necessary. The following scheme has been found to be satisfactory.

A stroboscope is used to obtain accurate angular velocity time data during starting and stopping. This is accomplished by observing the time at which the centrifuge reaches a speed preset on the stroboscope.

From Equation 4, it is apparent that the settling velocity at a given radius is proportional to the angular velocity squared. Therefore, if the angular velocity squared is plotted against the time during starting and stopping (Fig. 7), the area under the starting and stopping curves will be proportional to the distance settled during the respective times. The net correction is, therefore:

$$\text{Net correction} = \frac{\text{Area A} - \text{Area B}}{Y \text{ (to same scale as T)}}$$

#### CALCULATION OF RESULTS

Calculation of results from the sediment height *vs* time data is simple. The percentage of particles by weight larger than a given size is obtained by dividing the sediment height at that size by the total sediment height. For example in Table 1 the percent larger than 12 microns is

$$(8.3/26.1) \times 100 = 31.8 \text{ percent}$$

The percent undersize, or the frequency distribution may be obtained in the usual manner from the percent larger than distribution.

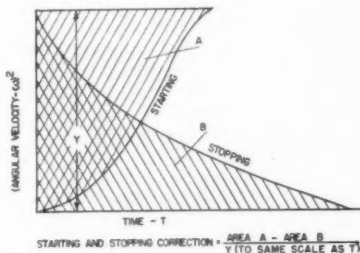


FIG. 7. TYPICAL STARTING AND STOPPING CURVES

## TYPICAL DATA

Fig. 3 illustrates a typical particle-size analysis obtained with this method as compared to the standard Andreason pipette and the size analysis supplied by the makers of the dust. At this time it will merely be stated that reproducibility and agreement with other methods have been found to be entirely satisfactory for all but a few unusual situations. Considering the compromises that have been made in the development of the method, agreement with other methods is good.

## ACKNOWLEDGMENT

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## DISCUSSION

W. C. L. HEMEON, Pittsburgh, Pa., (WRITTEN): The author is to be complimented on the ingenious apparatus and technique he describes for the determination of particle sizes. I wish to offer some comments on the general philosophy of particle size determination, comments which are directed primarily to the author's introductory remarks where he refers to the air filtration problem allowing the implication that particle size determinations employing dispersion in liquid may be applicable to the definition of size properties of dust in atmospheric suspensions.

In whatever field of technology, information on particle size distribution is only a means to an end. It is never of value for its own sake alone. It always serves merely as a bridge to the ultimate information which may range from pigment hiding power, reaction rates of pulverized material, the silting of rivers, the explosibility of organic dusts or the filtration of polluted air. In all of these instances the particle characteristics, using the term loosely, are a primary property having a direct bearing on the ultimate phenomenon. In practice, however, other qualities are frequently superimposed on those due to the ultimate particle size alone. Thus, for example, in liquid suspension freedom or presence of flocculation has a major influence on the sedimentation characteristics of particles and in the field of air cleaning similar problems are seen. Results of a particle size distribution measurement by a technique involving dispersion of the dust sample in a liquid may not be at all applicable to a study of the behavior of that dust as it passes in an atmospheric suspension through a filter. The small particles of carbon in a test dust mixture, for example, may very well fail completely to become separated from attachment to a larger particle even on passage through a compressed air jet; whereas, the dispersion occurring on suspension in liquid for the sedimentation test is likely to be complete. The technique described in this paper well might be useful in air filtration studies, but it should not be regarded as a factual description of the effective size of the dust in that operation.

W. A. CRANDALL<sup>a</sup>, New York, N. Y., (WRITTEN): The determination of particle size distribution in the subsieve size range has become an increasingly important problem in recent years. Dr. Whitby and his associates are to be congratulated for their work in this field.

We are keenly interested in this problem for a very important reason. We have spent, and are continuing to spend, millions of dollars to obtain the best equipment for removing fly ash from our boiler flue gases. The physical, chemical and electrical properties of fly ash are important factors in the design and operation of this equipment. Unfortunately, however, there has been a lack of uniformity and standardized procedures for determining these characteristics. This is particularly true in respect to the determination of particle size distribution. Therefore, we have undertaken, both on our own and in cooperation with other utilities, equipment manufacturers and research organizations, the task of evaluating existing size analysis procedures and developing other procedures that can be standardized throughout the industry. Hence, I was very glad to have this opportunity to learn more about Dr. Whitby's method and to comment on it.

There is one point that I would like to make before commenting specifically on various items in Dr. Whitby's paper. Most methods for the determination of particle size distribution are actually methods for the determination of terminal velocity. Dr. Whitby's method falls into this category. The terminal velocity distribution de-

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terminated by means of these methods is converted into particle size distribution by assuming that all the particles are spherical in shape and are homogeneous in respect to specific gravity. In many cases, it is the terminal velocity distribution in a particular gas or liquid that is actually desired. In our case, it is the terminal velocity distribution in flue gas that is required. Terminal velocity distribution determined in one liquid or gas can be converted to a terminal velocity distribution in another liquid or gas only if the condition of uniform specific gravity is satisfied, and into a true particle size distribution only if both the condition of uniform specific gravity and spherical particle shape are satisfied. Therefore, it becomes very important that one realize exactly what information about a subsieve size material is needed and what information the methods under consideration actually can give.

The outstanding characteristic of Dr. Whitby's method, in my opinion, is its ability to determine *particle size distribution* in the smaller size ranges in far less time than is required by conventional liquid sedimentation methods.

However, in studying Dr. Whitby's paper, there were several points, rather technical in nature, that I felt required further explanation. Perhaps Dr. Whitby would care to comment on these points later. These points were:

1. Can this method be legitimately extended to include particle sizes as small as 0.05 micron as stated in the paper? Shouldn't the Brownian Motion and possibly Cunningham's correction factor be considered in the determination of sizes smaller than 1 micron?

2. Only a few milligrams of sample are actually used in each centrifuge tube. What method is used to insure that this small increment is representative of the total sample?

3. In the calculation of results, the assumption is made that a given volumetric distribution in the capillary tube is directly proportional to weight distribution. When heterogeneous materials are segregated in accordance with terminal velocity, can it be assumed that a given volume of material at one point in the capillary tube has the same weight as an equal volume at another point in the capillary?

4. Do volumes of material observed during the period of normal settling become compacted during the periods of centrifuge operation, thereby causing an error in subsequent volume observations?

5. Considerable time has been devoted to the problem of corrections for the starting and stopping characteristics of the centrifuge. By the use of appropriate stroboscopic equipment, would it be possible to use continuous centrifuge operation, reading the volume of material in the capillary during the flash of a strobe light operating on a frequency equal to the rpm of the centrifuge? This should eliminate many of the calculations and corrections now required and shorten the time required for an analysis.

In conclusion, I again wish to commend Dr. Whitby for his work in the development of this method and congratulate him upon his presentation of his paper.

BRUCE L. EVANS, St. Louis, Mo., (WRITTEN): First of all I would like to congratulate Mr. Whitby on the very excellent work which he has done in connection with the measurement of small particles.

It seems to me that there are three divisions of the air cleaning technique. One of these is the determination of the contaminants in the air of large industrial areas so that a determination can be made of what must be removed by filters in any kind of heating, ventilating or air conditioning system.

Two, after samples have been collected it is necessary to have a simple method of determining the percentage of various sized particles to determine what is required of the filter.

Three, the design and testing of standard filtration equipment to determine its effectiveness in removing air contaminants so as to provide the ultimate consumer with the proper apparatus to give him the cleanliness required.

Based on the general description and operation of equipment made and tested by this paper, there is a very good possibility of developing simplified testing equipment that will be accurate enough for the normal commercial range required. This paper

should prove a valuable contribution toward correlating the studies on air cleaning requirements for human comfort.

S. SYLVAN, Louisville, Ky., (WRITTEN): We are all very interested in Dr. Whitby's work on particle size analysis. At the present time there are at least four methods more or less commonly in use and several different apparatus. None of them fills the requirements of scientific exactitude and practicable operation. One method may be convenient in some special field while a different method is preferred in others. Even within the small field of Air Cleaning we have to date been unable to accept a common method. Results are in disagreement and everybody seems to be using his own micron.

The main requirement of a dust analyzer is that it must be consistent with itself. If we produce a fraction that is supposedly less than, for instance, 10 microns a repeated test on this fraction must show that this is essentially so. It would be interesting to find how the Whitby apparatus performs in this respect. If it is equivalent to other methods I can see no objection against any details of the techniques employed.

The Whitby method may prove to be a very useful tool to determine an atmospheric dust condition. However, when it comes to defining the solids in atmospheric pollution it cannot furnish more than half the answer. We must differentiate between dust and smoke as two different and entirely independent constituents of the normal atmosphere. The entire problem has been aptly stated by Dr. Hemeon in a recent pamphlet, "Current Conceptions on Air Pollution". Either one or both of the constituents may be objectionable in different applications. Different types of equipment are available for combatting them and different measuring methods have been devised.

The size-frequency curve may be an adequate definition of the dust constituent but it is doubtful if the smoke constituent can ever be described in the same fashion. Particles of colloidal size are simply not accessible for this type of measurements.

We hope that Dr. Whitby and the Society will devote some attention also to this second phase of the pollution problem. For the art of air conditioning both phases are equally important.

R. S. FARR, Los Angeles, Calif.: I would like to compliment Dr. Whitby on the development of an instrument that I think is going to do a lot to further the air cleaning industry. This new instrument will provide a better means of determining air filter performance than the industry has had in the past. With this type of information a more practical study of our atmospheric dust problems and their solution will be possible.

Last week in Detroit the California Research Corporation gave a paper on the effect of particle size on engine wear. Research of this type is constantly going on and Dr. Whitby's instrument, which allows a quick method of particle size determination, certainly ties in to this kind of research and will provide a method for studying the effectiveness of air cleaning devices.

C. S. LEOPOLD, Philadelphia, Penna.: This excellent paper deals with the analysis of a sample as collected on or in a sampling device. In order to use this method to determine filter performance, it is necessary that we know whether the particles arrived at the sampling device as discreet particles or as agglomerates. It is not sufficient to know the size distribution after the sample is broken down. I should like to ask the author what has been done on that phase of the problem.

AUTHOR'S CLOSURE (Doctor Whitby): I am in complete agreement with Mr. Hemeon's view that particle size analysis is only one of a number of characteristics of airborne dusts that are of interest in the air cleaning problem. I also realize that the size distribution measured while the particles are dispersed in a liquid may be quite different from the distribution of aggregates actually existing in the airborne dust. However, size analyses on a great variety of bulk dusts as well as considerable numbers of recent analyses on airborne dusts sampled in our laboratory have shown that by proper technique it is possible to obtain size distributions by sedimentation which are close enough to the distribution existing in the air to be useful for many purposes.

The usefulness of size analysis results by any method must always be judged in the

light of the end application and of the known characteristics of the method. Many of the characteristics of this method are discussed in Part II on Procedures and Applications. Other characteristics, limitations and uses will be discussed in future papers on specific applications.

Mr. Crandall's discussion is greatly appreciated and I have the following observations.

The Cunningham correction is necessary for particles below  $0.1 \mu$  in size settling in a gas. However, the mean free path in a liquid is so much less than in a gas that the correction is negligible above  $0.01 \mu$ . The effect of brownian motion is proportional to the ratio of the diffusion velocity to the settling velocity. Even though the particle size is small, the settling velocity is high at the speeds used in the centrifuges. Stokes' law is actually used in ultra centrifuge work well below  $0.01 \mu$ .

Some data on sampling is presented in Part II. The refinement necessary in the sampling procedure will, of course, depend on the accuracy desired in the final results. Where the most representative sample is required, a bulk sample of several pounds is blended and split down to several grams. Then this small sample is dispersed in liquid and an aliquot taken from this suspension for the size analysis. Except for large particles above about 70 microns, which shouldn't be run by this method anyway, this procedure gives as representative a sample as is necessary.

The third and fourth points in Mr. Crandall's discussion are taken up in detail in Part II of this paper.

One user of the method has actually equipped a centrifuge with a stroboscopic device for continuous reading. While there may be a few applications that would justify the extra complication and expense of such a reading method, a survey of the present applications of the method has shown that it probably is not justified. The virtues of the present simple reading system can best be appreciated by using the equipment for a short time.

Research sponsored by the T.A.C. on Air Cleaning of the ASHAE, toward the objectives outlined by Mr. Evans is actually in progress at the University of Minnesota at the present time.

Dr. Sylvan's comments regarding the lack of standardization among the users of particle size analysis methods is most appropriate. This need has been one of the factors guiding the design and development of this method. Although this method is a liquid sedimentation method we feel that its speed and versatility will give it a wide sphere of usefulness.

The distinction between a dust and a smoke as referred to by Dr. Sylvan is not clear to me. The particle size analyzer makes no distinctions between the various sources of particles by itself. However, a nicely stratified sedimentation column is obtained from which it is often possible to draw some interesting conclusions by observation of the color of the various layers of sediment. Such observations on samples of atmospheric dust indicate that the finest particles in an airborne dust may not always be the carbonaceous smokes. In numerous instances the black carbonaceous material has existed in the form of lacy aggrates having a Stokes' equivalent diameter from 1 to 3 microns. In these cases the finest particles formed a light brown layer on top of the black layer. Up to the present, the centrifuge method has proven to be useful on atmospheric dusts down to about 0.25 microns, regardless of the source of the particles. Development of suitable techniques by comparison with the electron microscope should extend its usefulness to even finer particle sizes.

It may be of interest to know that the method is being used by at least one corporation for applications similar to the one mentioned by Mr. Farr.

Concerning the discussion of Mr. Leopold, as is stated earlier and in a paper to be published, there are methods by which it is possible to estimate the state of aggregation in an aerosol. We have made satisfactory progress in this direction. However the highly technical nature of these methods precludes their discussion here. This work will be covered in a later report or paper.

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## EVALUATION OF PANEL-TYPE AIR CLEANERS BY MEANS OF ATMOSPHERIC DUST

By H. A. ENDRES\*, W. T. VAN ORMAN\*\* AND R. P. CARTER, JR.†, AKRON, OHIO

**T**HE PROBLEM of atmospheric air cleaning is largely one of removing finely divided and highly dispersed carbonaceous matter, commonly known as soot or smoke. It is the finely divided carbon present in a normal industrial or city atmosphere that is responsible for most of the soiling of walls and furnishings in buildings. These highly dispersed and electrically charged carbon particles become deposited on such surfaces by thermal or electrical precipitation, or by sedimentation after flocculation, causing a substantial economic loss in cleaning, redecorating expense and soiled merchandise. In this respect smoke and soot should be classified with moths and corrosion as a public enemy. The term *spring cleaning* is a common household expression well known to every property owner and tenant, which usually signifies turmoil (dictionary definition: exhausting or distressing labor, disturbance, or confusion) and is never contemplated with joy. This can be avoided, or at least substantially reduced, by efficient air cleaning.

The carbonaceous smoke particles present in a normal industrial or city atmosphere, like smokes in general, are characterized by particle sizes below  $0.5 \mu$  (microns). A study of atmospheric dust concentration in 14 American cities made by the U. S. Public Health Service in 1936 showed that the average amount of suspended matter in the air during the winter months was 0.51 milligrams per cubic meter, or 815 particles per cubic centimeter, of which 65 percent by weight consisted of carbonaceous matter and 35 percent of ash. This ratio of carbonaceous matter to ash of nearly 2:1 becomes nearly 3:1 on a volume basis, so that the ash is really of minor interest to air conditioning engineers. In spite of this fact, there has been a tendency to under-estimate the importance of the small particle size and highly dispersed carbonaceous matter.

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According to the study made by the U. S. Public Health Service, the median size of the dust particles present in the normal city atmosphere during the winter months is  $0.58 \mu$  in diameter. The variation in particle size from city to city was found to be very small. Manufacturers of carbon blacks, who make smoke by the carload, usually refer to the surface area of their products in terms of acres per pound. Thus, carbon particles of the median size normally present in a city atmosphere would have a surface area of somewhat more than 0.5 acres per pound, which represents a high degree of soiling power.

According to Drinker and Hatch<sup>1</sup>, clean country air contains as little as 0.1 to 0.2 milligrams of dust per cubic meter, but the atmosphere in manufacturing towns may contain as much as 2.0 milligrams or more. The latter figure is considerably higher than the average given by the U. S. Public Health Service. Drinker and Hatch state that the air within a modern air-conditioned building should not exceed 0.2 to 0.3 milligrams of dust per cubic meter, regardless of the concentration outside. However, these two authorities concerned themselves with the health and hygienic aspects of dust and not its smudging or soiling power. An atmosphere containing 0.2 milligrams of highly dispersed carbon smoke per cubic meter would give good cause for spring cleaning. In a system where the air is recirculated as many as 100 times per day it would have ample opportunity to do a thorough smudging job. Small carbon particles have very low settling rates, move with the smallest air currents, and in a sense are part of the air and go wherever it goes.

The authors of this paper believe that the dust used in evaluating air cleaning devices should be representative of the dust which is to be removed in service, and in most instances this is finely divided carbonaceous smoke of high blackening power. This concept led to the development of the weight-smoke test described in a previous paper.<sup>2</sup>

In a paper by R. S. Dill<sup>3</sup>, in which a photometric method of testing air filters is described, it is stated that tests with atmospheric dust and smoke from a fuel oil flame were attempted but gave uniformly low efficiencies and were difficult to evaluate, but a variation of this method was found satisfactory for evaluating electrostatic dust precipitators. The present authors have pursued this test method further, and by making certain modifications and refinements, have arrived at a procedure that can be applied to the panel-type air cleaning devices.

#### PREPARED TEST DUSTS

The authors wish to correct a commonly held misconception regarding prepared test dusts. The dusts most commonly used, or presently being advocated, for evaluating panel-type air filters are composed of relatively coarse siliceous particles up to  $100 \mu$  or more in diameter, and 25 percent by weight, or less, of carbon in the form of carbon black or lampblack. The intimation is that the carbon in this mixture is free and becomes dispersed when the dust is injected into the air stream during the test. The authors have found that it is not possible to disperse carbon black by itself into an air stream by any of the means thus far proposed<sup>2</sup>. When the finely divided carbon particles are associated with coarse particles, as in a test dust, they do not become disassociated when injected into an air stream.

In a test dust containing relatively large siliceous particles and fine carbon

<sup>1</sup> Exponent numerals refer to Bibliography.

particles, the latter coat over and adhere to the surface of the former due to adhesion and adsorption. This can be shown by microscopical examination and by means of a simple demonstration. If equal volumes of two powdered materials of substantially equal particle size are intimately mixed, the color of the mixture will be influenced to the same degree by the components. That is, a mixture composed of black and white particles will be gray in color and the particles will maintain their distinctive characteristics. If, however, the black material is finely divided, like carbon black, and the white material is relatively coarse, like Arizona road dust, the particles of the latter become coated with the former and the mixture will be black. The carbon black particles are then no longer free and will be carried on the coarse particles in the dust stream and behave as an integral part of

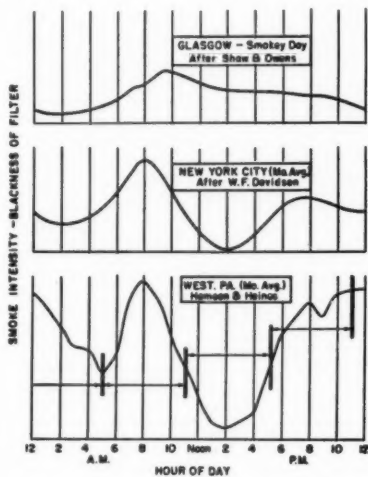


FIG. 1. HOURLY VARIATION OF SMOKE INTENSITY AT GLASGOW, NEW YORK AND WESTERN PENNSYLVANIA

the coarse particles. Therefore, such test dusts are not suitable for determining the efficiency of atmospheric air filters as they do not even approximate atmospheric dust.

The authors have shown in a previous paper<sup>2</sup> that a highly dispersed and finely divided carbonaceous smoke can be generated and controlled by burning mixtures of alcohol and benzene. This smoke can be used to augment or reinforce atmospheric dust in evaluating filter performance when the dust content of the air is low. This will be discussed later in the present paper.

#### HOURLY VARIATION IN ATMOSPHERIC SMOKE CONTENT

Referring to the work of Hemeon<sup>4</sup>, Fig. 1 shows the diurnal variations in smoke intensity in Glasgow, New York City, and Western Pennsylvania. It is significant

in these three radically different locations that a maximum occurs between the hours of 8:00 a.m. and 10:00 a.m. In this hemisphere a minimum of intensity occurs at about 2:00 p.m. The meteorologist will recognize that the time of minimum dust content, 10 a.m. to 5 p.m., coincides with the period of maximum wind velocities. The maximum smoke content coincides with the sunrise calm integrated with the starting of industrial activity for the day.

Through Hemeon<sup>6</sup>, the authors received curves showing the correlation between the smoke concentration in the air at Pittsburgh and Columbus on March 10 through March 31, 1954, as shown in the lower portion of Fig. 2. The Coh is a unit defined as the quantity of carbonaceous matter corresponding to an optical density of 0.01. Optical density is equal to  $\text{Log}_{10}$  (100/percent transmitted light). Hence, to express the number of Coh units, multiply the optical density by 100.

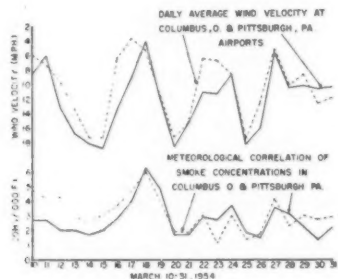


FIG. 2. METEOROLOGICAL CORRELATION OF SMOKE CONCENTRATIONS

This excellent correlation shows that the maximum intensity of smoke in the air occurs at the same time in these two cities. At the 1954 *Air Pollution Control Association* meeting, where these curves were presented, they aroused much interest. This was the first time that such a correlation had been demonstrated. A study of these curves indicates that a cyclic control factor, probably of meteorological origin, has caused these remarkable phenomena.

By plotting the daily average wind velocities at the airports at Columbus and Pittsburgh on an inverted scale, in the upper portion of Fig. 2, it is seen that the maximum smoke intensity occurs at a minimum wind velocity. Likewise, a minimum smoke intensity occurs at a maximum wind velocity. Hence the interpretation can be made that the smoke intensity of the air is inversely proportional to the wind velocity. This correlation between smoke content of the air and wind velocity should help clarify the authors' interpretation of the variables which are encountered in testing air cleaners with atmospheric dust.

Hubbard<sup>7</sup> has pointed out that the Coh is actually the number of square feet of dust deposit of optical density 1.0 per thousand cubic feet of air. An optical density of 1.0 means that the dust deposit will be 90 percent black. The authors believe that Hubbard's proposal of a standard of 25 percent reduction in transmitted light is to be preferred.

## IMPORTANCE OF PARTICLE SIZE

In his measurement of air pollution, Hemeon<sup>4</sup> pointed out that weight concentration measurements of dust in gaseous suspension give only the weight of the largest particles in a sample. This is true regardless of the absolute size of the largest particles. In outdoor measurements the weight concentration reflects only the air pollution nuisance commonly referred to as dust fall—the gravity deposition of solids on horizontal surfaces, which is primarily caused by large particles. If the pollution effect being explored is caused by small particles, weight measurements are worthless because the small particles contribute insignificantly to the total weight. This is illustrated in Fig. 3. Here it will be noted that if the one  $50\text{ }\mu$  particle size is captured, it will represent 97 percent of the total weight involved.

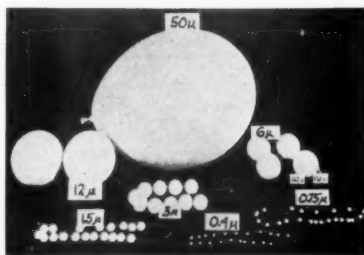


FIG. 3. MODEL OF TYPICAL OUTDOOR DUST DRAMATIZING EXTREME RANGE OF SIZES

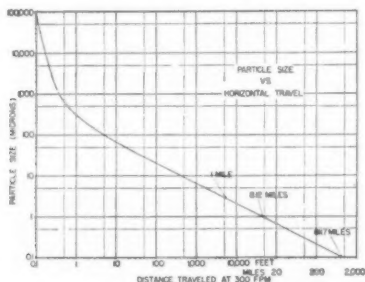
It is comparatively easy to capture a  $50\text{ }\mu$  particle and more difficult to capture the smaller particles. The weight approach is not adequate, for it gives practically no credit for the capture of finely divided carbonaceous matter.

Other authorities estimate that particles  $6\text{ }\mu$  and under represent over 99 percent of the soiling power of atmospheric dust. Herein lies the most difficult problem of air cleaning. It may be easier to understand why the smaller size particles are so difficult to capture by referring to Fig. 4. Here, the particle size in microns is plotted against horizontal travel in feet. The distance travelled horizontally is that distance through which the particle will travel in falling a vertical distance of 1 ft. For example, a  $100\text{ }\mu$  particle will travel only 5 ft. A  $10\text{ }\mu$  particle will travel about 430 ft. It is particularly significant to note that a one micron particle will travel 8.12 mi. In other words, there is an extremely rapid increase in the flotation power of the particles between  $10\text{ }\mu$  and the lower limits, which makes these particles incapable of capture by centrifugal or inertial forces. Only electrical forces operate effectively on them.

EFFICIENCY *vs.* PARTICLE SIZE

Begley<sup>6</sup> presented the curve shown in Fig. 5, which represents engine air cleaner efficiency plotted against particle size. This curve is similar to one published by a manufacturer of adhesive coated impingement filters, which shows efficiency

based on various particle sizes. The significant point shown by these curves is that below  $10\ \mu$  the efficiency of the filter falls off rapidly. This is characteristic of most impingement filters and clearly defines the area where greater effort in cleaning air should be directed.



NOTE: From data in p. 153, HEATING, VENTILATING, AIR CONDITIONING GUIDE 1953, Size and Characteristics of Air Borne Particulate Matter. All particles assumed spherical with density of 1.00 and constant horizontal velocity of 300 fpm. All distances represent horizontal travel during a fall of one foot.

FIG. 4. FLOTATION CURVE OF VARIOUS SIZE PARTICLES

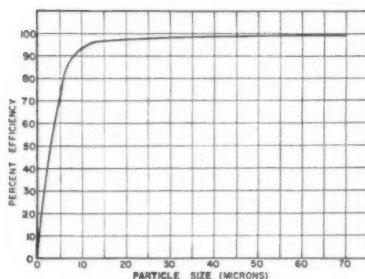


FIG. 5. ENGINE AIR CLEANER EFFICIENCY VS. PARTICLE SIZE OF THE DUST<sup>6</sup>

Fig. 6 shows the agglomerated carbonaceous material accumulated 30 ft downstream from a self-charging electrostatic air cleaner. These agglomerated particles range from  $150\ \mu$  to  $250\ \mu$  in size, whereas the initial carbonaceous matter in the air was of the order of one micron and less. The authors reason that if it is possible to cause agglomeration by such a simple mechanism, then by recirculating the air it should be possible to greatly increase the efficiency of the electrostatic air cleaners.

Therefore, they initiated a program of evaluating panel-type air cleaners by means of recirculating atmospheric dust, since most of the air is recirculated during the normal operation of heating and air conditioning systems.

The principles involved in the agglomeration of small particles have been demonstrated in the production of the furnace carbon blacks widely used in the rubber industry. The average particle diameter of such carbon blacks ranges from 0.001



FIG. 6. AGGLOMERATION OF CARBONACEOUS MATTER BY PASSAGE THROUGH AN ELECTROSTATIC AIR CLEANER, MAGNIFICATION 50X

to 0.5  $\mu$ . These compare closely in size with the carbonaceous matter found in the air. As Keevil<sup>8</sup> states, *Most of the airborne matter is in the size range of 0.2 to 1  $\mu$ , a micron being 0.001 of a millimeter or 0.00004 in.* Most furnace blacks<sup>9</sup> are made by a continuous process involving the partial combustion of natural gas or other hydrocarbons in specially designed furnaces and resulting in the formation of carbon black and gaseous products. After being cooled with sprays of water, the carbon black is separated from the gaseous stream by means of extremely high voltage in extensive electrical precipitator equipment, followed by a battery of tubular collectors. Fig. 7 shows such a system with the electrical precipitators turned on. It will be noted that no carbon black escaped, only a cloud of steam showing the effectiveness of the precipitation. A striking contrast is shown in Fig. 8, where the electrical precipitators have been turned off and clouds of carbon black escaped into the air. Thus, the principle of agglomeration is utilized in the collection of this fine, hard to catch carbonaceous matter. Of the carbon black used in 1952, 1  $\frac{1}{2}$  billion lb were collected by means of this mechanism.

In a previous paper on self-charging electrostatic air filters<sup>2</sup> the present authors showed that when two electrostatic filters are used in series the downstream unit may collect more dust than the upstream. At that time the authors were unable to explain this phenomenon, but since then it has occurred many times and there has been an opportunity to observe it more closely. Apparently, a particle which has escaped capture by an electrostatic filter acquires an electrostatic charge which causes it to agglomerate with other particles downstream. These ag-

glomerated and charged particles are more easily captured than the original particles. Hence, a downstream electrostatic filter often picks up more dust than the upstream filter.

#### EVALUATION OF PANEL-TYPE AIR CLEANERS WITH RECIRCULATED ATMOSPHERIC DUST

In the past it has been the practice to carry out evaluation tests of panel-type air cleaners on a single pass basis. By this procedure the air containing dust is



FIG. 7. SHOWING THE EFFICIENCY OF A CARBON BLACK PLANT WITH ELECTRICAL PRECIPITATOR AGGLOMERATING CARBON BLACK PARTICLES



FIG. 8. PLANT IN FIG. 7 WITH ELECTRICAL PRECIPITATOR TURNED OFF

passed through the air cleaner only once and then exhausted. As pointed out previously, this is not in accordance with normal air heating or cooling procedure where a major portion of the air is recirculated repeatedly. It is, therefore, logical to evaluate air cleaners by means of recirculated atmospheric dust and thus obtain

information regarding performance under actual service conditions. In an effort to accomplish this objective the following procedure was developed.

In order to run a continuous series of tests over a prolonged period without an operator present at all times, a heavy duty automatic dust sampler and recorder with special attachments was employed. This machine is shown in Fig. 9. A ribbon of Whatman No. 4 filter paper is passed through the mechanism and atmospheric air is drawn through the filter paper on a schedule of 25 min for a spot and then 5 min for moving the tape to the next spot. Automatic controls carry out this function so that continuous tests are conveniently made.

Whatman No. 41 filter paper is frequently used for atmospheric dust tests. Smith and Suprenant<sup>10</sup> have shown that there is less variation in Whatman No. 4 filter paper than in conventional Whatman No. 41. Visual examination shows more frequent occurrence of fine pinholes in Whatman No. 41 than No. 4. The rate of flow of the sample of air drawn through the paper is carefully controlled by maintaining constant suction.

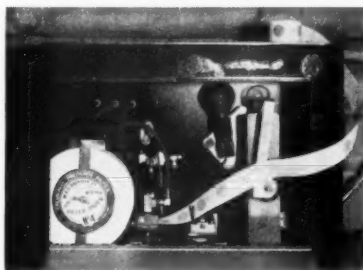


FIG. 9. AUTOMATIC DUST SAMPLER AND RECORDER

Recirculated air tests were conducted in a rectangular room having a volume of 2785 cu ft. The room has four windows and a door, all of which were sealed against air leakage with masking tape. A wind tunnel 16 ft long and 20 in. x 20 in. in cross section was used throughout the tests. In all the tests an air velocity of 300 fpm was maintained through the filter. Since this is equivalent to 833 cfm, the air in the room was passed through the filter every 3.34 min.

The tracing head inserts of the recorder were changed to produce a  $\frac{1}{4}$ -in. diameter spot and the filter paper tape drive speed was changed to 12 in. per hr. The vacuum pump was adjusted so that the vacuum was 5 in. Hg throughout the tests. Under these conditions a constant amount of air was sampled from the tunnel at a rate of 43.6 cfh. This amounted to 18.2 cu ft for each 25 min spot test.

A timer was used to supplement the recorder. The maximum setting of the timer was 120 min on and 120 min off; however, it was set so that air was sampled for 25 min and the filter paper tape was moved for 5 min—a total cycle of 30 min.

The air was sampled through  $\frac{1}{4}$ -in. diameter copper tubing from the upstream side of the filter being tested. This was done so that the dust level of the room air would be recorded and no immediate advantage taken of freshly filtered air.

Evaluation of the dust spots was made by means of a  $4\frac{1}{2}$ -in. 0-200 microammeter, a photocell unit, and a light source consisting of a 60-watt frosted glass bulb. The photocell unit consisted of a  $3\frac{1}{4}$ -in. diameter selenide coated photocell and a shield with a  $\frac{1}{2}$ -in. diameter hole in the center. When evaluating the dust spots, the tape was moved so that the spot fitted exactly over the hole in the shield and a reading was taken on the microammeter. The reading in microamperes was then converted into filter efficiency using the method described in the Appendix.

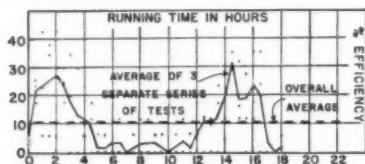


FIG. 10. RESULTS OF 107 ATMOSPHERIC DUST TESTS USING RECIRCULATED AIR AND 1 IN. VISCOUS IMPINGEMENT FILTER

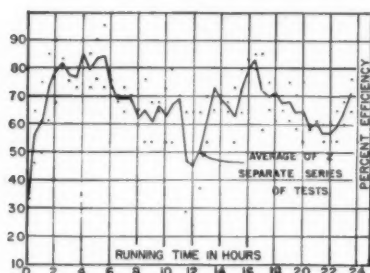


FIG. 11. RESULTS OF 95 ATMOSPHERIC DUST TESTS WITH RECIRCULATED AIR AND  $\frac{1}{2}$  IN. ELECTROSTATIC PANEL TYPE AIR CLEANER

In the initial test runs there was some air leakage into the room; however, this was later eliminated. The sampling pump for the recorder should not be located in the room where the tests are made, because the pump has an oil bath silencer and filter and oil droplets from the silencer may cause some discoloration of the filter paper.

#### EVALUATION OF VISCOUS IMPINGEMENT FILTERS

The performance of 1 in. viscous impingement filters, based on recirculated air tests, is shown in Fig. 10. The average efficiency by three separate series of tests,

or 107 individual tests, is 11.05 percent. This figure agrees with the commonly accepted values obtained by present test code procedures and published by filter manufacturers. It should be pointed out that in this type of filter no agglomeration takes place; hence, little or no benefits are derived from recirculation.

#### EVALUATION OF SELF-CHARGING ELECTROSTATIC AIR CLEANERS

Fig. 11 shows the results obtained with a  $\frac{1}{2}$  in. electrostatic air cleaner. Two separate series of tests (95 individual tests) gave an average value of 67.13 percent. This filter has a resistance of only 0.07 in. of water at 300 fpm air velocity.

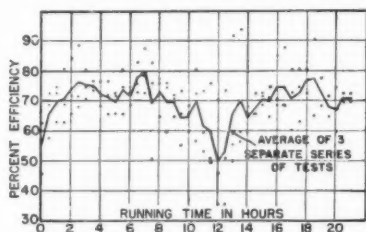


FIG. 12. RESULTS OF 122 ATMOSPHERIC DUST TESTS WITH RECIRCULATED AIR AND 1 IN. ELECTROSTATIC PANEL TYPE AIR CLEANER

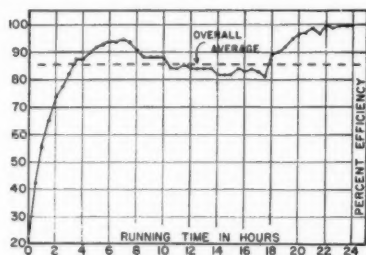


FIG. 13. RESULTS OF 48 TESTS WITH SMOKE-ENRICHED ATMOSPHERIC DUST AND RECIRCULATED AIR WITH 1 IN. ELECTROSTATIC PANEL TYPE AIR CLEANER

In Fig. 12 an average efficiency of 70.08 percent is shown for 122 separate tests on a 1 in. self-charging electrostatic air cleaner.

#### TESTING WITH SMOKE ENRICHED ATMOSPHERIC DUST

It has been shown that variations in the dust content of the atmosphere may make it difficult to test with this medium at all times. In order to meet situations

like this it has been found practicable to enrich the air with carbonaceous particles one micron and less in diameter by burning a mixture of 65 percent alcohol and 35 percent benzene in a small laboratory lamp for a period and then proceeding with the tests. Fig. 13 presents the data obtained with a 1 in. self-charging electrostatic air cleaner using smoke enriched air, covering 48 separate tests. The average value is 85.05 percent. The high efficiency values may be due to the fact that the smoke concentration in the air was considerably greater than normal. Reducing the time of burning the lamp should make it possible to obtain a normal smoke concentration in a given test room and thus duplicate the results of atmospheric dust tests. However, other factors may be involved, such as the magnitude of the charge on the freshly generated smoke particles. In commenting on a previous paper<sup>2</sup>, Nutting stated that *smoke generators produce highly charged aerosols, and charged aerosols are easier to collect than uncharged*. It has been our experience with self-charging electrostatic air cleaners, that the greater the dust content of the air, the higher the efficiency.

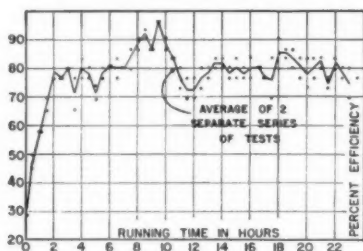


FIG. 14. RESULTS OF 93 TESTS WITH ATMOSPHERIC DUST AND RECIRCULATED AIR PASSED THROUGH TWO 1 IN. ELECTROSTATIC PANEL TYPE AIR CLEANERS SPACED 1 IN. APART

In Fig. 14, two 1 in. electrostatic air cleaners spaced 1 in. apart show an average efficiency of 77.79 percent in 93 tests. When several electrostatic air cleaners are employed in series, the benefits resulting from induced charges and agglomeration of the smaller particles by electrostatic attraction are clearly demonstrated.

In Fig. 15 is presented another concept of the effectiveness of two electrostatic air cleaners spaced 1 in. apart. Here the authors use the height of a column of air required to produce a 25 percent reduction in transmitted light through the filter paper as a criterion of efficiency. It will be noticed that at the beginning of the test the column of air was 8000 ft in height, the peak value was 250,000 ft, and the overall average was 42,000 ft. This is an excellent means of visualizing the dirt content of the air at any particular moment. The authors are greatly indebted to Hubbard<sup>7</sup> for the conception of this method of presentation and the mathematical calculations which follow.

$I_f$  = transmitted light, final.

$I_o$  = incident light through clean paper.

$L = (V_s/A_s)$ .

where

$V_s$  = cubic feet of air sample to obtain  $(I_t/I_o) = 0.75$ .

$A_s$  = spot size (square feet) =  $[\pi/(4 \times 144)] \times (1/2)^2 = 0.00136$  sq ft.

$V'_s = (25/60) \times 43.6 = 18.2$  cu ft when  $V'_s$  is cubic feet of actual air sample, and rate of flow is 43.6 cfh.

$$(V_s/V'_s) = [\log 0.75 / \log (I_t/I_o)].$$

therefore

$$L = [(V'_s/A_s) \times (V_s/V'_s)] = (18.2/0.00136) \times [\log 0.75 / \log (I_t/I_o)]$$

$$= (18.2/0.00136) \times [-0.1249 / \log (I_t/I_o)]$$

$$= (18.2/0.00136) \times [-0.1249 / \log (I_o/I_t)]$$

$$L = [1670 / \log (I_o/I_t)]$$

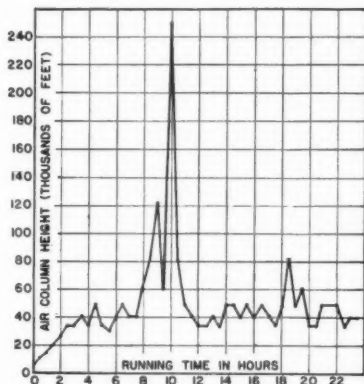


FIG. 15. PERFORMANCE OF ELECTROSTATIC PANEL TYPE AIR CLEANERS IN TERMS OF HEIGHT OF AIR COLUMN TO PRODUCE 25 PERCENT REDUCTION IN TRANSMITTED LIGHT. (RESULTS OF 47 TESTS USING ATMOSPHERIC DUST WITH RECIRCULATED AIR AND TWO 1-IN. ELECTROSTATIC PANEL TYPE CLEANERS SPACED 1 IN. APART)

Another method of illustrating the effectiveness of air cleaning devices, which was also originated by Hubbard<sup>7</sup>, is based on the decrease in the area of discoloration having 25 percent reduction in transmitted light. By this method it is possible to compute the cleaning benefits derived from the removal of soiling dust from the air. This is illustrated by Fig. 16. At the beginning of the test, the area of discoloration was 12 sq ft. This was reduced to a minimum of 0.4 sq ft during the test and an average value of 2 sq ft throughout the test.

## ACKNOWLEDGEMENTS

Photos through courtesy of W. C. L. Hemeon (Fig. 3) and Columbian Carbon Co. (Figs. 7 and 8).

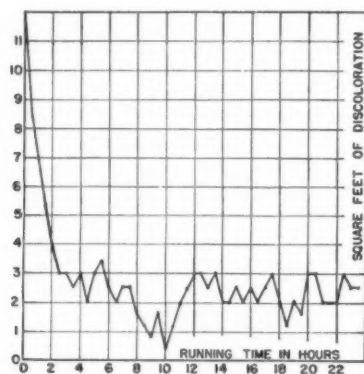


FIG. 16. DATA FROM TESTS IN FIG. 15 WHEN EXPRESSED IN TERMS OF SQUARE FEET OF DISCOLORATION. (25 PERCENT REDUCTION IN TRANSMITTED LIGHT)

## CONCLUSIONS

The problem of atmospheric air cleaning is largely one of removing finely divided and highly dispersed carbonaceous matter, commonly called soot or smoke. Therefore, air cleaning devices should be evaluated by means of atmospheric dust containing the normal concentration of soot or smoke.

Self-charging electrostatic air cleaners offer an effective and economical means of removing highly dispersed carbonaceous particles and other dusts normally present in the atmosphere, when used in conventional forced draft heating and air conditioning systems.

Passage of the highly dispersed dust and smoke particles through the electrostatic medium enhances the charge and causes agglomeration, thus increasing the air cleaning efficiency of a recirculating system.

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## APPENDIX

### EVALUATION OF THE DUST SPOTS

In order to evaluate the series of spots on the filter paper tape an equation has to be used which will allow the range of efficiencies to vary from 0 to 100 percent depending on darkness of the dust spot. Equations based on upstream and downstream spots are not applicable because the recorder registered only one spot for each test period. However, it is possible to derive an equation from the Lambert-Beer law of transmitted light which gives the desired data. This law can be stated mathematically as follows:

$$\log (I_0/I_T) = KCL$$

where

$I_0$  = incident light.

$I_T$  = transmitted light.

$K$  = a constant depending upon the transmitting medium.

$C$  = unknown concentration of sample.

$L$  = depth of the medium.

A few assumptions must be made before this equation can be applied. First, it is assumed that the microampere output of the photocell is directly proportional to the light falling on the cell. It is also assumed that the reflectance and diffusion of the light through the filter paper are constant for all dust spots. This is not exactly true, but the differences are so small that they can be ignored. It is also assumed that the light transmittance through the clean filter paper is constant, which assumption was checked and found to be true.

Since every spot on the tape is compared with the zero spot, the latter should be described first. This spot will be designated  $S_1$ . Any other spot compared to the zero spot,  $S_1$ , can be called  $S_2$ . The zero spot (dust spot sampled from the unfiltered atmosphere) can be designated by the equation:

$$\log (I_0/I_1) = K_1C_1L_1$$

and the spot sampled from filtered air can be designated by the equation:

$$\log (I_0/I_2) = K_2C_2L_2$$

In these equations:

$I_0$  = transmitted light through clean filter paper.

$I_1$  = transmitted light through dust spot from normal air.

- $I_2$  = transmitted light through dust spot from filtered air.  
 $C_1$  = normal dust level.  
 $C_2$  = unknown dust level of filtered air.  
 $K_1$  = a constant depending on the transmission at spot  $S_1$ .  
 $K_2$  = a constant depending on the transmission at spot  $S_2$ .  
 $L_1$  = thickness of the filter paper at spot  $S_1$ .  
 $L_2$  = thickness of the filter paper at spot  $S_2$ .

Dividing the equation for  $S_1$  by the equation for  $S_2$ ,

$$[\log (I_0/I_1)/\log (I_0/I_2)] = [(K_1 C_1 L_1)/(K_2 C_2 L_2)] \quad (1)$$

Reducing Equation 1,

since

$$K_1 = K_2$$

and

$$L_1 = L_2$$

$$[\log (I_0/I_1)/\log (I_0/I_2)] = (C_1/C_2) \quad (2)$$

Now solving Equation 2 for  $C_2$  (the dust level after filtration):

$$C_2 = C_1 \times [\log (I_0/I_2)/\log (I_0/I_1)] \quad (3)$$

Since  $C_1$  is the normal dust level and  $C_2$  is the dust level after filtration, the percent efficiency would be equal to:

$$[100 (C_1 - C_2)]/C_1 \quad (4)$$

Substituting Equation 3 into Equation 4 and reducing, a usable equation is reached, viz.:

$$\begin{aligned}
 \text{Percent Efficiency} &= \frac{100(C_1 - C_1 \frac{\log I_0/I_2}{\log I_0/I_1})}{C_1} = 100 \frac{C_1}{C_1} \left[ 1 - \frac{\log I_0/I_2}{\log I_0/I_1} \right] \\
 &= 100 \left[ 1 - \frac{\log I_0/I_2}{\log I_0/I_1} \right] \quad (5)
 \end{aligned}$$

The efficiency becomes zero when  $I_2 = I_1$ , which is the same as saying that the dust level of filtered air is the same as the dust level of the unfiltered air. The efficiency is equal to 100 percent when  $I_2 = I_0$ , or when the air contains no dust. These two conditions can be proved by simply substituting the values in Equation 5.

Equation 5 is actually the same as that of Hubbard<sup>7</sup>, viz.,

$$\text{Percent Efficiency} = 100 \left[ 1 - \frac{t_u \times Q_u \times \log \frac{\mu a^* \text{ downstream start}}{\mu a \text{ downstream finish}}}{t_D \times Q_D \times \log \frac{\mu a \text{ upstream start}}{\mu a \text{ upstream finish}}} \right]$$

Because each spot on the tape represents the same total flow of air, the  $t_u \times Q_u$  term equals the  $t_D \times Q_D$  term and therefore cancels out. The ( $\mu a$  downstream start) and

\*  $\mu a$  = microamperes.

( $\mu$ a upstream start) terms apply to clean filter paper, which is  $I_0$ . The ( $\mu$ a downstream finish) and ( $\mu$ a upstream finish) are the same as  $I_2$  and  $I_1$ , respectively.

The results obtained with the Dill dust spot tester, manufactured according to the National Bureau of Standards design, can also be related to this equation. Since the spots on the filter paper are balanced photometrically before, during, and after the test, the logarithmic terms cancel out leaving

$$\text{Percent Efficiency} = 100 [1 - (t_u \times Q_u)/(t_D \times Q_D)] = 100 [(t_D Q_D - t_u Q_u)/(t_D Q_D)]$$

Usually the time of sampling upstream and downstream air is the same, and the equation reduces to:

$$\text{Percent Efficiency} = 100 [(Q_D - Q_u)/Q_D]$$

The Dill spot tester is useful in exploring single pass efficiencies of panel air cleaners.

## DISCUSSION

LESLIE SILVERMAN<sup>†</sup> and D. M. ANDERSON,<sup>‡</sup> Boston, Mass. (WRITTEN): This article is interesting but also highly controversial. The use of atmospheric dust as a means of testing air cleaners is not new and has been used for many years at our Air Cleaning Laboratory and by many investigators before World War II. In fact, in the literature on air cleaner evaluation there is mention of the use of atmospheric dust as a testing suspension prior to 1930. The work to which the authors refer by the U. S. Public Health Service was actually done in 1931-33 and appeared in 1936. The work was that of Ives, et al<sup>1</sup>. A study<sup>2</sup> with regard to carbonaceous content of air previously reported showed that the amount of free carbon may vary from less than 10 to 50 percent but that 25 percent on a weight basis would be a representative mean value. The ash or the inorganic material is not of negligible interest because what the housewife or maintenance personnel wipe off furniture and floors is largely the settled dust which is coarser material. A large number of fines <5  $\mu$  (microns) are present along with the larger material and by number they predominate.

The soiling power of the atmosphere has been recognized by many filter test codes which have been developed such as the original ASHVE code and the present AFI code; both incorporate carbon in the mixture. The National Bureau of Standards also uses an ash which contains carbon for a test suspension. For many years atmospheric dust soiling power has been used by manufacturers of air conditioning precipitators as a means of rating performance. These are usually rated on a basis of 85 to 90 percent efficiency by stain on atmospheric dust. The authors' approach could hardly be considered as a new concept.

The question of recirculation is not disputed but it must be recognized that the average home with warm air heating system will have from 1 to 2 air changes per hour, whereas a modern air conditioned establishment might reduce this value to 10 to 25 percent make-up air, depending upon the conditions involved. Because of the large amount of disturbed settled dust which is drawn into the air in a house and that which is suspended by sweeping and other activities it cannot be stated categorically that the dust should only be carbonaceous. A representative dust should be one that simulates recognized atmospheric dust composition.

The chief reason for atmospheric dust not being used as a test medium for panel-type filters in the past has largely been because of its high variability. The major objections lie in the fact that the length of time per test necessary to evaluate the

<sup>†</sup> Air Cleaning Laboratory, Harvard School of Public Health.

<sup>1</sup> Ives, J. E., et al (*U. S. Public Health Service Bulletin No. 224, 1936*).

<sup>2</sup> A Partial Chemical Analysis of Atmospheric Dirt Collected for Study of Soiling Properties, by C. E. Moore, Robert McCarthy and R. F. Logsdon (ASHVE JOURNAL SECTION, *Heating Piping & Air Conditioning*, October 1954).

holding capacity of the filter at outdoor dust loads would be impractical. This is to be borne in mind since the filter for which the test method is proposed is one which has a low holding capacity as determined by our laboratory and as the authors demonstrate by the tests reported in this paper.

Another important difficulty with atmospheric dust, in addition to its variations in concentration, particle size, and composition, which have been recognized for many years, is the fact that these variations make it difficult to test from one moment to the next unless simultaneous sampling (that is, samples up- and downstream from the filter under test) is employed or a material balance should be made with absolute filters. The authors of this paper have apparently overlooked this important factor.

The method of enrichment which the authors have proposed and used is completely uncontrolled by their method of generation. It could only be practical if enrichment were controlled so as to produce a constant air concentration and composition in proportion to the air conditions existing at the time of test.

The authors' method of producing the carbonaceous smoke is actually not capable of producing any better dispersion than trying to disperse carbon black or lamp black directly with an ejector. A highly charged and chaining aerosol is produced by burning.

The difficulty of dispersing carbon black is because a high charge is generated or produced by contact which serves to reaggregate the carbon even if the cohesive force of the particles is overcome.

The authors have described the use of electrostatic precipitators for collecting carbon black as employed in practice. Actually, it should be pointed out that at the present time most of the carbon black produced in this country is collected by means of cyclones followed by bag filters or cyclones followed by electrostatic precipitators. The use of the electrostatic precipitator as an agglomerator is a fairly recent innovation in the carbon black industry but has been used in at least two power plants of which the authors have knowledge, (one installation of this type is 16 years old). The purpose of this approach is to reduce the size of the precipitator necessary by using it as an agglomerator. The authors neglect the fact that agglomeration is produced by applying a voltage of 50,000 to 80,000 volts in the collector. The fact that agglomeration takes place at 8 to 10 kv voltages is proved by the fact that several low voltage precipitators use a blow-off filter in series to catch agglomerated material which may be blown off the plates in use, if an adhesive is not used on the plates.

The importance of particle size in filter testing has been emphasized in work on air cleaning for many years. The concept of fractional particle guarantees dates back to at least the early 1930's. It is established practice in power plant specifications to use fractional guarantees for mechanical collectors. In any device, depending upon impingement or mechanical collection, including filters, the particle size becomes important. The important concept, however, is not size alone, but terminal velocity which incorporates size, shape, distribution, surface characteristics, and density.

It is interesting to note that in Fig. 6, the authors state that the agglomerated material ranges from particles 150  $\mu$  to 250  $\mu$  in size. Settling velocity of such particles is so high they would not stay suspended in the air for any length of time. For example, the settling velocity for a 250  $\mu$  particle of unit specific gravity is greater than 150 fpm. Thus, selective removal in the test room would take place.

The authors state that only electrical forces operate effectively on particles below 10  $\mu$ . However, impaction forces are effective on particles of 1  $\mu$  and less and below 1  $\mu$ , diffusion and thermal forces operate. In fact, the most effective filter we have today, which is capable of removing 99.98 percent of 0.3  $\mu$  particles and smaller, operates with minimal electrostatic effects. The sampling filter that the authors used as their method of standardization behaves similarly. The statement fails to recognize basic principles of aerosol filtration.

Using a continuous spot analyzer sampling upstream of the filter and not downstream, the authors show that the efficiency of an impingement filter rises as high as 25 percent in Fig. 10 and then drops to nearly 0 and again rises between 10 and 18 hr. If the room is completely protected against leakage how do they account for this second

rise in concentration? Furthermore, if over the period of several hours an impingement filter is operating at 20 percent efficiency it will have removed practically all of the dust in the room.

The important thing to recognize in the impingement filter coated with a viscous medium is that re-entrainment is unlikely because material caught on the filter is trapped by the adhesive and cannot escape as long as a sticky surface is available.

If the room is sealed, as indicated by the authors, calculations show that with the air sampler they employed and assuming that it is 100 percent efficient, approximately 30 percent of the room air volume is drawn through the sampling filter. Some of the efficiency of filters tested in this fashion would be influenced by the cleaning effect of the sampler itself. The room also acts as a settling chamber and the circulating fan can be an agglomerator also. The sampling pump was discharging air from the room and since it removed  $\frac{1}{3}$  of the air volume of the room over its sampling period it is obvious that at least  $\frac{1}{3}$  volume leakage was present or else a reduced pressure would result.

The authors mention that Smith and Suprenant have shown there is less variation of #4 filter paper than conventional 41. An examination of their article reveals the fact that on atmospheric dust, #41 filter paper has 26.5 percent efficiency by count whereas #4 filter paper is only 15 percent. On DOP both filters give about 23 percent efficiency. On  $0.5 \mu$  particles the comparison between the two papers is almost identical whereas on  $0.6$  to  $0.8 \mu$  size particles the #41 filter paper is 64 percent efficient by count and the #4 paper is only 38 percent efficient. In our opinion this does not show #4 is a better paper. Both papers are made by the same process and the avoidance of numerous pin holes in either type of paper can only be done by pre-selection.

The authors make the point that their self-charging filter is an agglomerating device. Evald Anderson<sup>3</sup> many years ago stated that filters are agglomerating devices. Perhaps we should eliminate those filters coated with a viscous medium because any attempt to retain the particles prevents their subsequent contact by reentrainment. Anderson showed that by passing dusty air through a bed of fibers a large number of agglomerates could be produced in the effluent air. We can show from the authors' data that dust is apparently re-entrained from their filter which they contend is superior for atmospheric dust. Agglomeration, if it exists, can be due to re-entrainment as much as to electrostatic charge. Referring to the actual multiple data tests they state that two 1 in. self-charging filters in series show an average of 77.79 percent (Fig. 14) on atmospheric dust compared to 70.08 percent (Fig. 12) for a single filter. Calculation from these data would indicate that the improvement of two in series is 7.71 percent and based on the difference between 7.71 percent and the penetration through a single filter of 29.92 percent, this makes the second filter 26 percent efficient. We believe this shows conclusively that agglomeration by the first filter does not exist to any appreciable extent. The authors' first pass efficiency data for the so-called self-charging filter confirm tests made in our laboratory and many other laboratories indicating efficiencies on the order of 10 to 15 percent which falls rapidly and is highly variable due to the low holding capacity.

If the room were sealed, as indicated by the authors and re-entrainment did not take place, why would not the air in the room become 100 percent clean within a short time? An equation which assumes that the fractional removal for each pass is the same through such a filter can be derived readily. Assuming that the dust concentration in the room is  $10^7$  particles per cubic foot of air with the number of air changes used by the authors, the efficiency necessary so that the dust concentration is reduced to 1 particle per cubic foot at the end of 24 hr is

$$1 = 10^7 (1 - E)^{24 \times 60 / 3.31}$$

The efficiency of the filter to produce this change in concentration over the 24 hr period

<sup>3</sup> *Chemical Engineers' Handbook*, Second Edition, McGraw-Hill Book Co., 1954.

need only be 0.4 percent. This, of course, neglects the fact that the sampling filter has also removed  $\frac{1}{4}$  of the air in the room as well as the room settling effect.

It is obvious then, that the authors' room was not tight and that re-entrainment takes place continually during a test to show a continual performance of their filter. The test on the viscous impingement filter would indicate that the dust concentration in the room is reduced because the dust caught is held on the filter and therefore the total dust level has changed.

We cannot exactly interpret the authors' computation of efficiency. If the original dust concentration in the room was used for the first measurement and all efficiencies are based on this value one result is obtained. If efficiencies are calculated on a basis of the initial value before the filter and then the test value on the next run used for the difference another value is obtained. In either case, the cumulative efficiency rises rapidly. If the original dust level is used the cumulative efficiency value rise is not as rapid.

We did not have an opportunity to comment on the authors' original paper with regard to self-charging filters but we have tested this filter in our laboratory as have many others with no evidence of any charge being generated on the filter. We have generated a charge on such fibers by mechanical friction and then tested filter performance and again we have not demonstrated any continued effect on a normal dust load in atmospheric air cleaning. In our tests clean air passing over the fibers does not create a charge at the velocity used in filtration. We would like to ask the authors, in view of the principle of conservation of energy, how they can explain the *self-charging* performance of their filter? Where does the energy come from to produce the charging? The dust level in normal air does not contain enough particulate matter to cause charging by friction against the fibers as measured in our tests. In heavy dust loadings friction charging can take place but these loadings would be in the range of process effluents or at least 1000 times atmospheric dust loadings.

If the particles in the atmosphere already contain a charge and are deposited on the filter this charge might be available but this would also be a factor in other dielectric media rather than metal filters where the charge might leak off. However, the amount of charge which the dust could hold under such conditions would be minimal and by our measurements not of significance.

We regret that we cannot concur with the authors' contention that this filter is self-charging with atmospheric loading and the velocities employed. As a mechanical filter we find that it fails to provide adequate holding capacity. The authors' tests have conclusively proved, in our opinion, that re-entrainment is taking place. The other bases for tests which the authors mention, namely, the use of height of air column and square feet of discoloration seem to offer nothing but another pair of arbitrary bases for investigators.

A. B. ALGREN and K. T. WHITBY,<sup>□</sup> Minneapolis, Minn., (WRITTEN): The statements in the opening paragraph and in other places in the paper, that the problem of air cleaning is largely one of removing the finely divided carbonaceous matter, is an over simplification. There is abundant evidence that any air cleaner for residential use must be designed to cope with the lint problem. Furthermore, recent particle size studies at the University of Minnesota indicate that residential dusts may contain substantial amounts by weight of very low terminal velocity fragments of fiber that have a Stokes' equivalent spherical diameter in the 3 to 10  $\mu$  size range. The settling velocity of these particles is low enough so that enough of them to be a considerable nuisance could stick to vertical surfaces.

The quoted figure of 65 percent carbonaceous matter in atmospheric dust seems rather high. Recent studies by McCarthy and Moore<sup>4</sup> indicate an average free carbon content of about 35 percent for five different samples.

<sup>□</sup> Research Associate. Mechanical Engineering Dept., University of Minnesota.

<sup>4</sup> Determination of Free Carbon in Atmospheric Dust, by Robert McCarthy and C. E. Moore (*Analytical Chemistry*, February 1952, p. 411).

Also there is considerable evidence from dispersion studies at the University of Minnesota that a considerable portion of the free carbon in atmospheric dusts adheres to the larger dust particles under normal conditions. During sedimentation size analysis synthetic mixtures of carbon and other dusts such as fly ash and lint can always be separated in clean layers with the larger particles appearing bright and clean. However, for atmospheric dusts the particles in all but the largest size ranges appear dark gray or black.

Recent work by Dalla Valle, Orr and Hinkle<sup>5</sup> indicates that the aggregation in an aerosol is caused primarily by turbulence, diffusion and thermal effects. Normal electric charges on the particles affect the shape of aggregates of particles but not the aggregation rate. This is not true in the strong electric field of electrostatic precipitators where ions are present.

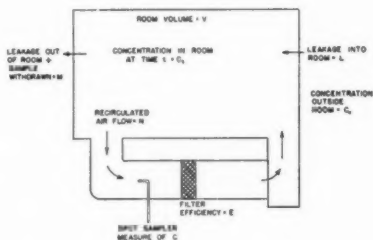


FIG. A. DIAGRAM OF TESTING ARRANGEMENT

The source of the particles shown in Fig. 6 is not clear. Were these from an enriched air stream or are they normal atmospheric dust particles? What did the particles at this same station look like without the electrostatic air cleaner in the air stream?

A study of the methods and data given on the evaluation of panel type air cleaners casts doubts on the validity and usefulness of the proposed methods. There are three serious objections to the methods proposed.

The first of these is the basing of the efficiencies of a test covering a period of 24 hr on a dust spot sample taken at the beginning of the test. The wide fluctuations in atmospheric dust concentration possible are clearly visible in Fig. 1. These fluctuations undoubtedly account for the wide fluctuations in apparent efficiency visible in Figs. 10 to 16.

The second and third objection can be best illustrated by a nonsteady state analysis of the data given in Figs. 10 through 14. Fig. A of this discussion shows the system described by the authors.

Rowley and Jordan<sup>6</sup> derived an equation from which it is possible to calculate the dust concentration in a ventilated and filtered room. This equation is:

$$C_t = \left[ C_0 - \frac{LC_0 + D + FC_0(1 - E)}{M + NE + Av} \right] e^{\frac{-(M + NE + Av)t}{V}} + \frac{LC + D + FC_0(1 - E)}{M + NE + Av} \quad (D-1)$$

<sup>5</sup> The Aggregation of Aerosols, by J. M. Dalla Valle, C. Orr, Jr. and B. L. Hinkle (*British Journal of Applied Physics*, Supplement No. 3, 1954).

<sup>6</sup> Predicting Dust Concentration, by F. B. Rowley and R. C. Jordan (*Engineering Experiment Station, Technical Paper No. 26*, University of Minnesota, 1942).

where

$C_t$  = concentration of dust in the room at any time  $t$  after the start of the filtration process.

$C_i$  = initial concentration in the room at  $t = 0$ .

$C_o$  = concentration of dust in the air outside the room.

$L$  = infiltration air — cubic feet per minute.

$M$  = exfiltration air — cubic feet per minute.

$N$  = recirculated air — cubic feet per minute.

$F$  = outside ventilation air — cubic feet per minute.

$E$  = filtering efficiency.

$V$  = room volume — cubic feet.

$A$  = room floor area — square feet.

$v$  = effective settling velocity of dust particles — feet per minute.

$D$  = dust concentration produced inside per minute.

For the situation represented by Fig. A, ( $L = M$ ) ( $F = 0$ ), ( $D = 0$ ) and it may further be assumed that the fine particles measured by the spot test have no appreciable settling velocity, ( $v = 0$ ). Under these conditions, Equation D-1 becomes

$$\frac{C_t}{C_o} = \left[ \frac{C_i}{C_o} - \frac{L}{L + NE} \right] e^{-\frac{(L+NE)t}{V}} + \frac{L}{L + NE} \quad \text{D-2}$$

From Figs. 10 through 14, it will be noted that the apparent efficiency builds up during the first several hours. From this first part of the efficiency curves shown in the figures and from the equilibrium efficiency reached, it is possible to evaluate the unknowns in Equation D-2. In Equation D-2, there are three unknowns  $C_i$ ,  $L$  and  $E$ . For the case where atmospheric dust is used as the test dust  $C_i = C_o$ . The two remaining unknowns  $L$  and  $E$ , can be solved for in the following manner.

Take logarithms of both sides of Equation D-2, and obtain:

$$2.30 \log \left[ \frac{\left( \frac{C_t}{C_o} \right) - \left( \frac{L}{L + NE} \right)}{\left( \frac{C_i}{C_o} \right) - \left( \frac{L}{L + NE} \right)} \right] = -\frac{(L + NE)}{V} t \quad \text{D-3}$$

It will be noted that this is the equation of a straight line on semi-log papers and is the equation that describes the first part of the efficiency curves of Figs. 10 through 14. In these figures, the ordinate called percent efficiency is actually  $[1 - (C_t/C_o)]$ . The quantity  $L/(L + NE)$  in the left hand member of Equation D-3, can be obtained from Equation D-2, for large values of  $t$ .  $L/(L + NE)$  is the value of  $C_t/C_o$  obtained after the dust concentration in the room has reached an equilibrium. It corresponds roughly to the overall average efficiency values given in the papers for each figure. However, some judgment had to be exercised in selecting  $C_t/C_o$  values from the curves to be

TABLE A

(1)	(2)	(3)	(4)	(5)
$t$ MIN.	% EFF. $\left\{ 1 - \frac{C_t}{C_o} \right\} \times 100$	$\frac{C_t}{C_o}$	$\frac{C_t}{C_o} - \frac{L}{L + NE}$	COL. 4 $1 - \frac{L}{L + NE}$
30	47	0.53	0.33	0.416
60	57	0.43	0.23	0.288
90	67	0.33	0.13	0.162
120	78	0.22	0.02	0.025
150	77	0.23	0.03	0.0375
180	80	0.20	0	0

used in the calculations because of the erratic nature of efficiency curves as a function of time. Using the value of  $L/(L + NE)$  obtained in the stated manner and  $[1 - (C_t/C_0)]$  values taken from the curve it is possible to calculate values of the left hand members of Equation D-3 for various times as shown in Table A. If these values are now plotted against time on semi-log paper as in Fig. B it will be seen that they fall roughly on a straight line from the origin at  $t = 0$  and  $\alpha = 1$ . The slope of the line drawn from the origin through these points is actually the quantity  $(L + NE)/V$  in Equation D-3. Simultaneous solution of these two equations yields the value of  $L$ , the infiltration rate into the room, and  $E$  the actual single pass efficiency of the filter.

The following calculation for the data of Fig. 14 illustrates how this is done. Columns 1 and 2 Table A are the data taken from the curve. From column 2 the data of column

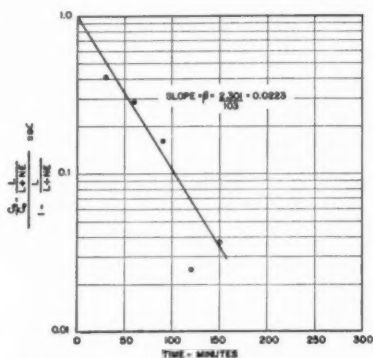


FIG. B. VALUES FROM COLUMN 5 OF TABLE A PLOTTED AGAINST TIME

5 were calculated. Column 5 is then plotted as shown in Fig. B. Next a straight line is drawn from the origin through the first several points and the slope determined. Then:

$$(L + NE)/V = 0.0223. \quad (D-4)$$

where

$$\begin{aligned} V &= 2785 \text{ cu ft.} \\ L + NE &= 2785 \times 0.0223 = 62 \text{ cfm} \end{aligned}$$

From the equilibrium conditions

$$L/(L + NE) = 0.20 \quad (D-5)$$

Solving Equations D-4 and D-5 it is found that:

$$\begin{aligned} L &= 12.4 \text{ cfm} \\ E &= 6.0 \text{ percent} \end{aligned}$$

The results from similar calculations based on the data provided in the other figures are tabulated in Table B.

From this table two things are apparent. First, the infiltration of air into the room

is high enough so that this factor rather than the filter efficiency is the most important factor in determining the apparent efficiency values shown in Figs. 10 to 14.

TABLE B

FIGURE	FILTER	DUST	L CFM	E %
10	1 in. Viscous Impingement..	Atmospheric	60	2.8
11	½ in. Electrostatic.....	Atmospheric	11.2	6.6
12	1 in. Electrostatic.....	Atmospheric	19.3	8.2
13	1 in. Electrostatic.....	Smoke Enriched	10.0	3.6
14	Two 1 in. Electrostatic.....	Atmospheric	12.4	6.0

Second, it can be seen from Equation D-2 that the apparent efficiency as shown in the figures will become 100 percent if the leakage is actually reduced to zero. Under these conditions any filter could theoretically give 100 percent efficiency if it was allowed to run long enough.

Also the data of Table B show that the actual single pass efficiency of all the filters tested is below 10 percent as would be expected. Even if liberal allowances are made for the difficulties in taking data from the curves and allowances are made for the effects of the assumptions made, it is doubtful if the single pass efficiency of any of the filters tested is above 15 percent.

*The Spot Test:* Calculation of the face velocity through the filter paper used in their sampler gives a value of 533 fpm. Studies at the University of Minnesota show an average efficiency by the spot method on atmospheric dust of about 50 percent for filter papers at this face velocity. Furthermore, it has been found that the efficiency varies widely from day to day as the composition of the atmospheric dust varies, especially at face velocities above 150 fpm. These two factors will introduce a further uncertainty into the significance of the data given in the paper.

Where accurate results from the spot test are desired, it is advisable to use a high efficiency filter material such as glass fiber mat or millipore filters. If the indicated or similar papers are used, they should be used at face velocities below 100 fpm and even then the efficiency should be checked occasionally by backing with a high efficiency material.

R. S. FARR, Los Angeles, Calif. (WRITTEN): This paper presents a very interesting concept of panel filter evaluation. The authors are to be congratulated on the work that has gone into securing the data for this paper and the excellent organization of the paper.

It is agreed by practically all persons working with air filters that the fine particles in the air are very largely responsible for the smudging observed. It is also known that the fine particles are the most difficult to catch by any means. The desirability of testing air filters of all kinds with a dust representative of atmosphere pollution has been recognized for some years. The authorities quoted by the authors are outstanding but other data not referred to, collected by such laboratories as the Bell Telephone Laboratory, show that the character and particle size distribution of airborne matter varies widely over the United States. This fact has discouraged many efforts to establish a test dust that truly represents all the airborne material that would approach air intake filters. This known variability makes it most difficult to compare, on the basis of atmospheric dust, one filter with another unless the two filters are tested in a number of locations scattered over the United States. For purposes of comparison, therefore, it seems reasonable to adopt a standardized test dust that will give data comparable to that shown in Fig. 5 of the paper in which performance is plotted against particle sizes in microns. With a curve such as Fig. 5 available to the user and a knowledge of local dust conditions, performance of the filter in place could be predicted.

With reference to the particles,  $6\mu$  and smaller, the authors state that they represent 99 percent of the soiling power of atmospheric dust. Work currently being undertaken by the Technical Advisory Committee on Air Cleaning may make possible the determination of filter performance of dusts in this particle size range without having to depend on day-to-day variability of atmospheric dust. Further with respect to these fine particles, the authors state that they remain airborne for long periods of time and, therefore, unless they are electrostatically activated or otherwise efficiently impinged, they should not constitute a soiling problem to merchandise or horizontal surfaces in air conditioned spaces. Facts gathered by the writer by means of adhesive coated microscope slides placed on exposed surfaces in department stores confirmed this fact in that there were no particles under  $5\mu$  observed on any of the 40 slides used. The slides were exposed for 8 to 10 days and were found to be covered with lint and a few large dust particles. Recirculated air in the space tested would have been loaded with lint and a recirculated air filter test based on the authors' proposed method would not have properly evaluated the filter for this application.

The curves of Figs. 10, 11, 12, 13 and 14 present interesting data. In the appendix to the paper, the authors state that the efficiency plotted as ordinate on these curves is determined by comparing dust spot blacknesses between the spot taken at the beginning of the test and each consecutive spot in turn. On this basis the constant efficiency of two or more consecutive  $S_2$  spots of the same density indicates that no dust was removed from the air during that particular period of the test. A consecutive series of  $S_2$  spots of decreasing density would show as an increasing efficiency. A consecutive series of  $S_2$  spots of increasing density, indicating an increase in the dust concentration of the air ahead of the filter in the authors' test apparatus, would show as a decreasing efficiency on the curve. All of these conditions are present in the curves of Figs. 10 through 14 at one place or another so that careless interpretation of the curve by one unfamiliar with the details of the testing method could lead to erroneous conclusions with respect to the performance of the filter.

To make the foregoing point clear, the efficiency curve of Fig. 11 rises steadily during the first two hours indicating a continuing removal of dust from the recirculated air. From approximately hour 4 to hour 12 the curve slopes downward indicating either that the filter unloaded or that additional dust was introduced into the air stream in some other fashion. From hour 12 to hour 16 the efficiency curve rises in general indicating that the filter was again collecting dust. From hour 16 to hour 22 the curve sloped downward indicating an increase in dust concentration in the recirculated air. Since the authors state that the room was very carefully sealed, it may be that vagaries in the air currents within the room stirred up additional dust from time to time, but a test method subjected to such uncontrolled factors appears to lack the consistence of results.

The function of air filters is to remove particulate matter from the air. Recirculated air applications subject filters to artificial dusts of many kinds and concentrations. While the test proposed by the authors may provide valuable information, lack of knowledge of the dust being collected makes the test comparative for one location only.

F. LANDGRAF, McKees Rocks, Pa., (WRITTEN): The authors have described a means of evaluating air cleaner performance in a closed system by periodically recording the dust content in the recirculated air represented as a proportionate function of the initial dust content.

Since the recirculating system is sealed and no additional dust could enter after the run began, it is difficult to understand the negative slopes shown on the curve in Fig. 10. This curve would lead one to believe that the viscous impingement filter had twice loaded up to where it had retained approximately 25 percent of the dust in the system and twice unloaded until after 17 hr it was perfectly clean and the dust content of the system was the same as when the run began.

This seems inconceivable, since with no filter in the system at all, based on the *floatation curve* shown in Fig. 4; after 17 hr the air would have traveled about 58 mi at a

maximum velocity of 300 fpm (much slower in the room) and, therefore, should have *dropped out* an appreciable portion of its dust load.

Obviously, dust *fall out* becomes a factor in any closed recirculating system. However, this factor is at best very difficult to evaluate and bears no relation to the filter under test. Therefore, a method of evaluating a filtering device including an unrelated indeterminate factor is open to question.

ARTHUR NUTTING, Louisville, Ky. (WRITTEN): In discussing this paper, I want to refer to Fig. 11. Anything I say about this figure can be applied also to Figs. 12, 13, and 14 which report test results on synthetic fiber filters. In considering this test, we should keep in mind that we are recirculating air from an air tight room at 15 air changes an hour. A filter is in the air circuit. The room is sealed tight so dust cannot enter by accident and no dust is added in any way to the air system.

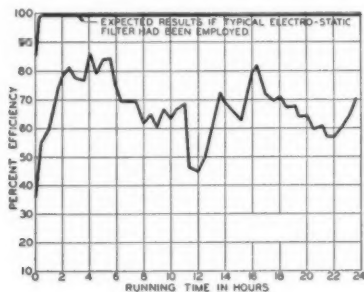


FIG. C. EQUIVALENT OF FIG. 11 OF  
AUTHORS' PAPER

In this Fig. 11, the authors have measured the amount of dust in the air at regular periods and have compared the measurements to the quantity of dust that existed at the beginning of the test. This curve then, expresses the dust condition in the room throughout the period of test and is not an expression of filter performance. Fig. 11 does not show directly what filter efficiency is—so I would like to take the authors' data and interpret it in the only way I think it makes sense. (See Fig. C.)

Referring to Fig. C at the first test point, 35 parts of dirt have been removed—65 percent (the quantity above the line) remain in the air. At the 2-hr interval—78 percent of the dust has been removed and 22 parts (the quantity above the line) of the original quantity are left in the air. Now, let us look at the dust condition at the 12 hr point. At this point, the air cleanliness (filter efficiency as the authors call it) there were only 22 parts at 2 hr. After 10 hr of operation, during which time the air and dust passed through the filter about 150 times, the air is  $2\frac{1}{2}$  times dirtier. Now this is a test of air filter performance, and if we credit the filter for improvement in cleanliness, then we must also charge it for making the air more dirty. So, I believe it is correct to say that during this 10-hr period, the filter effectiveness is minus 150 percent. Going to the end of the test; at 24 hr, the air cleaner contains 30 parts of the original dust whereas at the 2 hr interval, it contained 22 parts. So, in 20 hr operation, the net result in room condition, is an addition of 8 parts of dust, or a minus efficiency of 35 percent. Minus because the air is dirtier.

Now, I believe it is correct to use the condition of the air at the 2 hr point as a base of comparison for the following reasons:

1. It is well known that dust particles will naturally agglomerate and by gravity settle out. This will make the air cleaner.
2. Air turbulence in a fan system will accelerate this natural agglomeration and cause more than normal dust settling, thus resulting in cleaner air.
3. Dust will impinge on all the surfaces and remain there and the authors make reference to this in their opening paragraph. This will make the air cleaner.
4. By thermal precipitation, dust will deposit on all surfaces. All these natural phenomena will act to make less dust in suspension. So, for these reasons, in a recirculation system, the room air will immediately begin to become cleaner as soon as the fan system is started and would do so whether or not an air filter were in the system.

This test method is not acceptable, but if it were, the authors should have operated the system for a period of time to allow the dust condition to come to a balance before beginning their test, and thus, eliminate the just mentioned variables. By using the

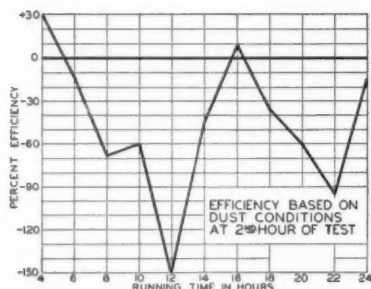


FIG. D. EFFICIENCY BASED ON DUST CONDITIONS AT SECOND HOUR OF TEST

condition of air at the 2 hr point interval on Fig. C, we eliminate the variables and the filter performance can be expressed more accurately. Now, I want to show you point by point how the complete curve would look with the 2 hr point used as a base.

The filter (See Fig. D) has only two points of positive efficiency because there are only two points where the air is cleaner after the second hour. The performance falls to -150 percent at the twelfth hour as explained before.

Now the efficiency can be expressed in another way, a way more usual for an engineer. That would be by expressing the efficiency in terms of the hour to hour change. For instance, by comparing the condition at the fourth hour with the second and the condition at the eighth hour with the sixth, and so on. This will give a performance curve for this filter as shown in Fig. E. There are as many negative points as there are positive, which agrees with the authors data, and it should be this way because there was practically no change in the cleanliness of the room after the second hour. Now I do not believe this filter added dust to the air at any time. In my opinion, the air was made dirtier during the test because the test room was not air-tight as the authors believe. I think that the dust measurements as shown in Fig. C are really a reflection of changing dust conditions in the outside atmosphere—and, if this is so, the test has no integrity.

At the 1952 Annual Meeting of the Society in St. Louis, the authors presented a paper

on this same type of self-charging electrostatic filter. Since that time, some one-half dozen similar filters have been placed on the market. Air filter manufacturers have had an opportunity in the past two years to investigate all of these synthetic fiber filters and I know that they will support the statement that there is absolutely no electrostatic effect present in this type of air filter. None of us has ever been able to measure an electrostatic charge on the filters. We have tested them in every conceivable way and they do not remove smoke or in any way perform like an electrostatic filter. We have even applied electrostatic charges of 15,000 volts from the external

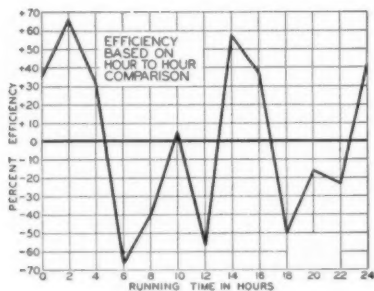


FIG. E. EFFICIENCY BASED ON HOUR TO HOUR COMPARISON

power supply and cannot measure that the filter cleans air to any degree except in the first few minutes of operation, there may be a few percent of dust removal. If the fiber of this type filter becomes slightly coated with dust or the relative humidity raises to the order of 50 percent, the surfaces become electrically conductive and even with the external power supply, the charge will disappear and the dust removal becomes zero. If an unlimited source of energy does not force the filter to perform, then how can it be expected that air friction in an air conditioning system can do so?

The data expressed in Fig. C, therefore, when interpreted as it should be, support the findings of air filter manufacturers that filters of this type made of synthetic fibers have absolutely no electrostatic properties in an air filter.

These questions naturally came to mind:

1. The authors object to synthetic dusts on the ground that the small carbon particles ride piggy-back on the coarse particles. I believe this is what happens in nature; little particles join each other and little particles ride on big ones, and I ask if the authors can produce any evidence to the contrary.

2. We are acquainted with the fact that charged dust will precipitate heavily on the walls of a ventilated space. If this device charges the atmospheric dust, what would prevent the dust from being collected in the room never reaching the filter in succeeding air changes?

3. The authors have had several years to test these filters in every way and they have even devised new test methods, but what is the filter efficiency as measured by the usual methods of test such as: U. S. Bureau of Standards Method, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS Filter Test Code or tests by the *Air Filter Institute*.

B. L. EVANS, St. Louis, Mo. (WRITTEN): This paper is well written and certainly goes to the heart of evaluating the performance of the various types of panel type air

cleaners and their effectiveness as applied to the removal of the atmospheric dust which is to be found in the average industrial area.

From the results shown on various types of panel air cleaning equipment it is very obvious that some cleaning equipment is very low in efficiency, especially in the smaller sized ranges. This study correlates with the present study being conducted by the Technical Advisory Committee on Air Cleaning and will, I believe, eventually lead to the development of a testing code so that the end user will be able to determine in advance what results he may expect from the type of filtering equipment chosen.

This paper will help point up the usefulness of present filtering equipment and help in the final determination of what a user has a right to expect when buying air cleaning equipment.

G. W. PENNEY, Pittsburgh, Pa. (WRITTEN): This paper includes a discussion of *Testing with Smoke Enriched Atmospheric Dust* and discusses it particularly as related to the testing of what the authors call *self-charging electrostatic air cleaners*. If we are testing an air cleaning device to determine the efficiency at which such a device will clean air in service, I would like to question the use of anything except the actual atmospheric air pollution. It is my opinion that the only ultimately conclusive test is one made on the air cleaner as installed in service and tested on the actual atmospheric pollution on which it must operate. Fortunately, the air pollution in most cities is sufficiently similar so that a test in one city is frequently indicative of the performance in another city. However, artificial smokes may be entirely different from normal atmospheric pollution and so, in my opinion, they should not be used for performance tests unless it can be shown that they behave like the actual atmospheric pollution to be removed.

The use of artificial smokes or dusts seems particularly questionable in the case of *self charging electrostatic* devices. Since such devices are made from high resistivity plastic materials, they can, of course, hold an electrostatic charge for a time. This may temporarily provide an electric field for precipitating charged particles, but there does not seem to be any adequate means of replacing the charge which leaks off or is used up by precipitating charged dust. There is likewise no means for charging the dust particles entering the filter.

Since the device does not provide for charging the dust entering the filter any charge on the dust as it enters the filter is vitally important. Freshly dispersed dusts usually have a strong electric charge. Likewise freshly generated smoke particles may be charged. However, atmospheric dust is very weakly charged if charged at all. James W. Hansen in his doctor's dissertation at the University of California (1948) measures the initial charge and rate of loss of charge on dust. He shows that even though the particles polluting the atmosphere may have had a charge as generated, when suspended in the atmosphere for a time these particles would lose their initial charge through the natural ionizing processes in the atmosphere.

The paper refers to *Testing with Smoke Enriched Atmospheric Dust*. In enriching atmospheric dust the smoke added usually becomes the controlling factor. My experiments indicate that benzene-alcohol smoke has unusual characteristics. Using the mixture specified in the paper and passing the smoke at low velocity through the collector cell of a two-stage precipitator with voltage on the plates but no charging voltage, this benzene-alcohol smoke could be efficiently removed. Normal atmospheric dust would not be efficiently removed under these conditions. This demonstrates the initial charge on the smoke indicating that it is not a suitable material for testing these so called *self charging* devices which have no provision for charging the dust.

Another peculiar characteristic of benzene-alcohol smoke particularly for a mixture of 50 percent alcohol and 50 percent benzene is the tendency to form chains when passing through an electrostatic field. Of course, many dusts show a slight tendency to form chains under these conditions, but the benzene alcohol smoke seems to show it to a degree that is entirely unlike normal atmospheric dust. This chain formation

could explain the improved performance on recirculation since these long chains would be readily removed by any filter.

In conclusion, it is my opinion that one must be particularly careful in the testing of these filters made of plastic materials. It seems reasonable that one can readily devise a test using freshly generated smokes or dust which are initially charged and likewise handling the filter in such a way that the plastic is also charged. A high efficiency might then be obtained for a short time. However, it is my opinion that in normal operation any initial charge on the plastic will either leak off or be neutralized by collecting ions from the air so that after a short time the device must function as an ordinary filter. Any test must, therefore, accurately represent actual operating conditions in all particulars if it is to be conclusive.

W. K. GREGORY\*, Louisville, Ky., (WRITTEN): My conclusion after reading this paper, is that the authors started out with the untenable premise that the prime function of an air filter is to remove particles ranging in size from 0 to 0.5  $\mu$ .

The authors state that, *the median size of the dust particles present in normal city atmosphere during the winter months is 0.58  $\mu$  in diameter.*

I do not propose to question this statement, but after 31 years' experience in the air filter industry, I know for a fact that the major portion of the dust which reaches an air filter consists of much, *much* larger particles. When I entered the air filter business in 1924, the only filters which were manufactured at that time were 4 in. thick, made of steel wool or metal ferrules and coated with oil. Similar filters, somewhat improved, are in wide use today and they get good and dirty in a relatively short time in most localities.

These viscous impingement filters are the foundation on which the air filter industry was built and, as one of the many manufacturers of such filters, our company is free to admit that the efficiency we obtain on dust particles ranging from 0 to 0.5  $\mu$  is relatively low, possibly not much more than the 11.05 percent shown in Fig. 10 of this paper.

Nevertheless, with a good viscous impingement filter, you can eliminate the greater portion of the dust which would otherwise settle on walls and furniture, and for the average home or office such filters, while far from perfect, have proven adequate.

The high concentration of dust above 0.58  $\mu$  in size which is caught by viscous impingement filters is probably due to the fact that the outside air which enters them is usually taken in close to a roof or sidewalk, where winds or passing vehicles throw relatively large particles in the air. Where air is recirculated, large particles are placed in suspension by people walking about and tracking in dirt from the outside.

It is, therefore, imperative that in devising a test procedure for the evaluation of air filters, that larger dust particles which are capable of being airborne, be included in any test dust used.

*The Air Filter Institute*, which includes all the pioneer companies in the industry and whose members account for at least 80 percent of the dollar value of all air filters sold, has given a great amount of thought to the development of an air filter test code by which the practical value of unit air filters may be determined.

The test dust chosen has the following analysis: 25 percent K-1 lamp black, 3 percent No. 7 cotton linters, and 72 percent Standardized air cleaner test dust (fine).

Particle sizes for the K-1 lamp black range from 0.01 to 0.5  $\mu$  and fine air cleaner test dust contains 39 percent of particles in the 0 to 5  $\mu$  range, so we have a total of 53 percent of particles under 5  $\mu$ . The largest particles in this test dust are 80  $\mu$ , and only 22 percent of the particles are larger than 20  $\mu$ . According to the AFI code procedure, this dust is picked up from a traveling tray at a uniform rate by an air actuated aspirator and is dispersed into the duct ahead of the test filter under 60 lb air pressure. The dust fed to the filter is weighed and the dust passing the filter is caught by an absolute filter having an efficiency of 99 percent plus, after which it is weighed. Efficiency is thus accurately determined on a weight basis.

\* Continental Air Filters, Inc.

Granted that some agglomeration of the particles of lamp black does occur, and some coating of the larger particles of air cleaner test dust takes place, it is, nevertheless, evident from comparative tests with other types of dust, that the lamp black is broken up into very fine, hard-to-catch particles.

As an example, every filter we have tested has shown a lower efficiency on *AFI* test dust than on standardized air cleaner test dust (fine) alone. The air cleaner test dust contains 39 percent by weight of particles in the 0 to 5  $\mu$  range and the *AFI* dust contains only 28 percent of particles of this size plus 25 percent lamp black. This indicates that a considerable portion of the agglomerates which undoubtedly occur in the lamp black are effectively broken up by air pressure employed for dispersing the dust.

In conclusion, I would like to say that, while it is certainly desirable to remove as large a percentage as practical of dust particles in the 0.5  $\mu$  range, no air filter is practical which will not show a reasonably high efficiency when tested by the *AFI* method. Since our company manufactures a filter which employs synthetic fibers and could, therefore, be considered a *self charging electrostatic air cleaner* (although we have not used this nomenclature), I am not disposed to minimize the value and practicability of filters of this general type, for commercial reasons, if nothing else. Nevertheless, our company has taken the position that to be practical, a filter of this type should show an efficiency of at least 70 percent when tested in accordance with the *AFI* code and should have a reasonably high dust holding capacity.

LESTER T. AVERY, Cleveland, Ohio (WRITTEN): This paper on air cleaning is most interesting. Those of us who use air conditioning for ourselves and supply it to others will agree with the premise *smoke and soot should be classified with moths and corrosion as a public enemy*.

Well do we recall the early disappointments in the forced air heating system with the much-extolled filter to find we still had discoloration above grilles and continued need for paint-washing and paper-cleaning. It seems that the authors' explanation of how soot collects on more heavy particles which can be filtered out adds a little light to the question of why you get so much discoloration downstream from a filter. It appears to me that the soot may be riding along on the heavier particle, sort of a loose skin effect. This is the only rational explanation we have for the very obvious advantage of the electrostatic cleaner in holding this black material.

This paper also gives me an opportunity to comment on the basic fallacy of thinking on the part of the ordinary warm air heating industry. Somehow, somebody decided that fan operating cost was important so fans were designed with a minimum pulling power and the smallest possible motors and then put on intermittent operation. Nothing could be more disastrous to good air cleaning. If we somehow could get the manufacturers, contractors and owners to understand the problem, we would use heavier fans and quiet, put on larger motors and plan for continuous fan operation, thus getting the virtues of the recleaning that is so ably pointed out in the paper. Even a filter that is only 10 percent efficient on small particles will gradually clear them up by the repetitive process of passing through 6 or 8 times an hour and limiting the new material to what comes in by infiltration or brought in on rubbers and shoes and clothes. So I submit that, in making better filters, we must have stronger fans to handle air against higher resistances.

It also permits me to point out that ordinary air cleaning is not simple, should not be over-simplified in its selling or equipment. Good air cleaning may involve stage operation whereby we have different means to remove gases, heavy dirt and smoke and, of course, odors that do not necessarily come out with filtering the above. "Cleanliness is next to Godliness" and I think it is as true in air as it used to be in people.

Now that our Society is using a new name, AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., we are taking a very serious interest in air conditioning problems and air cleaning is certainly one of them. Again thanks to the authors for this paper.

W. C. L. HEMEON, Pittsburgh, Pa. (WRITTEN): It is difficult to evaluate the significance of the recirculated air efficiency data presented in this paper because so many details of procedure and results have been omitted. The basis for evaluating the solids content of a particular filter paper sample is essentially the same as the blackness test and is basically sound, provided, however, that the electrical output of the photocell is, in fact, directly proportional to the incident light. This cannot be assumed to be true as the authors indicated was done, otherwise entirely false conclusions from test data may result where photocell and ammeter are incorrectly matched.

The mathematics in the appendix do not tell us clearly how the efficiency data were obtained from a sequence of single blackness measurements on filter paper samples all taken upstream of the air cleaning unit. It is necessary to infer that the procedure consisted of the comparison of one 25-min sample with a similar sample immediately following it, and that the ratio of the corresponding optical densities of the two samples provided the basis for calculating efficiency during that hour.

If our inference is correct we are then led to a question illustrated by the following example. In the absence of any reported test data let us assume an initial concentration of solids during a particular 25-min period of 6 COH's per 1000 ft of air, a fairly smoky atmosphere<sup>7</sup>. Taking 70 per cent as a typical efficiency figure, we would expect smoke concentrations to be reduced in about 3 hours to a level where the accuracy of the light transmission reading is seriously impaired. We are, therefore, unable to understand how it was possible to obtain satisfactory measurements for periods up to 24 hours in length unless outdoor polluted air was admitted to the test chamber at intervals. Perhaps the authors failed to emphasize that the atmosphere was enriched with the artificial smoke early in the test period. In that event one might suggest that enhanced efficiencies were due to the particular character of the artificial smoke.

No evidence is reported in the paper to prove that there was no significant infiltration of outdoor air which, if it occurred during the periods, would subject the test results to the characteristic variability of diurnal changes in outdoor smoke concentrations. The scatter of data shown in the graphs, in particular the dips seen after several hours have elapsed, do, in fact, suggest the possibility of such influences.

Apart from the content of this particular paper I should like to comment on the filter testing controversy that has occupied the attention of a segment of the Society for several years. In a consideration of outdoor air pollution<sup>8</sup> due to dust and smoke we have found it useful in clarification of the problem to make a sharp distinction between the properties and effects of coarse particles which are characterized by the ability to settle out by gravity and fine particles like fuel smoke which do not. We see no reason why the same considerations do not apply to the cleaning of air for building ventilation, and if it were recognized it would logically lead to the conclusion that there are two types of dirt to be considered—coarse dust and fine smoke. The soiling properties of each are entirely different, their origin is not the same and the difference in their size properties indicates that they cannot be measured by the same technique. From this it follows that two separate and distinct test methods are needed. The blackness test, one form of which was applied by the authors in this paper, would represent an appropriate test method for one type of soiling potential, and a weight efficiency method would be applicable to the other type. It would then be of paramount importance always to emphasize in applications that the two types of pollutant are as different in all respects as, figuratively speaking, a gas and a liquid.

C. B. ROWE, Madison, Wis. (WRITTEN): In my opinion, the basic philosophy which guided the preparation of this paper is well summed up, in a previous writing,<sup>9</sup> by one of the authors . . . *our philosophy is that if you can pick up the finer particles, coarser*

<sup>7</sup> Determination of Haze and Smoke Concentrations by Filter Paper Samplers, by W. C. L. Hemeon, G. F. Haines, Jr., and H. M. Ide (*Air Repair*, August 1953).

<sup>8</sup> Hemeon, W. C. L., Senseshaugh, J. Deane, and Haines, George F. Jr., *Measurement of Air Pollution, Instruments*, Vol. 26, April, 1953.

<sup>9</sup> Self-Charging Electrostatic Air Filters, by W. T. Van Orman and H. A. Endres (*ASHVE TRANSACTIONS*, Vol. 58, 1952, p. 76).

particles will take care of themselves. In the work described in this paper, it appears that the coarse particles are left to take care of themselves. Any work which completely ignores the existence of the larger particles is of limited value.

The terms *atmospheric air* and *atmospheric dust*, from general use in the field and from the authors' paper, apply to air and its dust load as found in the out of doors.

The majority of filters that are manufactured and sold are used in such applications as home furnaces, unit heaters, and central system filter banks. Air passing through filters used for such applications is generally 75 to 100 percent re-circulated. Only from 0 to 25 percent of the volume circulated is what is referred to as *atmospheric air*; 75 percent or more of the air is *re-circulated air*.

For purposes of discussing air filtration, I define *re-circulated air* and *re-circulated air dust* as the air and its dust load as it occurs in space used for general occupancy.

To gain some conception of the nature of the dust load in *re-circulated air* we removed a number of filters from various types of installations such as home furnaces, package

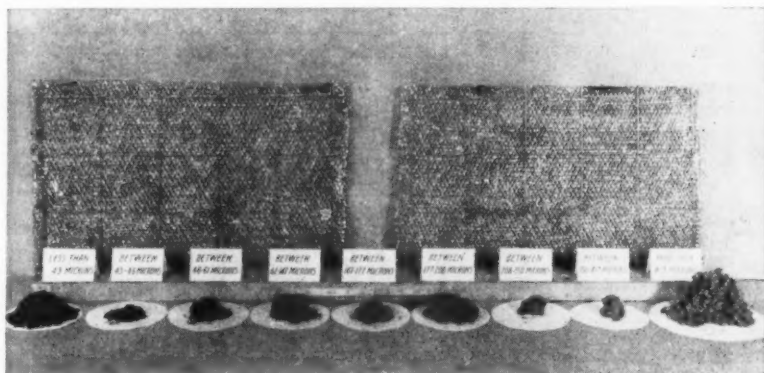


FIG. F. SIZE AND QUANTITY ANALYSIS OF DUST FROM TWO FILTERS IN AN APARTMENT HOUSE FURNACE

air conditioners, and central system filter banks. We washed the dust load out of each of these filters with dry cleaning fluid and then collected the dust load from the cleaning fluid by filtration. The components of the collected dust were then separated by sieving. A typical analysis is shown in Table C. Fig. F shows a typical set of extracted dust specimens.

The foregoing procedure can be criticized on the basis that, (1) particles which the filter did not arrest are not accounted for, and (2) particles which the filter did arrest possibly agglomerated and thus increased the apparent particle size. However, from visual and microscopic examination of the dust particles it was very apparent that a substantial part of the total bulk was made up of particles which are vastly larger than those contained in the *atmospheric air* described by the authors of this paper. For example, all of the samples checked contained substantial quantities of lint. In our opinion, it is dust of this type which panel type filters are commonly used to collect, and in selecting a test dust, dust of this type certainly should be considered and included. We believe that the significance of the dust load in *atmospheric air* is, at most, of secondary importance.

We recognize the importance of recirculation. However, it is certainly not realistic to go on from there and place a filter testing unit in a completely sealed and presumably

unoccupied room. Such a test cannot duplicate the conditions present in an occupied room. Test results, particularly if air flow resistance changes are included, obtained in an unoccupied room are of no value in predicting the performance of an air filter in an occupied room. This is especially true of the conditions of an average home involving rugs, furniture, pets, clothing, children, and opening and closing doors.

Under the heading Evaluation of Viscous Impingement Filters the authors state that efficiency values of 11.05 percent were obtained on viscous impingement type panel filters. They further state that this value is in line with the commonly accepted values obtained by present test code procedures and published by filter manufacturers. The

TABLE C—SIEVE ANALYSIS OF DUST SPECIMEN EXTRACTED FROM A PANEL FILTER USED IN A DEPARTMENT STORE CENTRAL SYSTEM FILTER BANK

SIZE OF FRACTION IN MICRONS	WEIGHT OF FRACTION IN GRAMS	WEIGHT OF FRACTION IN PERCENT OF TOTAL WEIGHT	NATURE OF FRACTION
+417	7.3	8.1	Lint, Coarse dust, Agglomerated particles.
-417 +350	1.9	2.1	Lint, Coarse dust, Agglomerated particles.
-350 +208	3.3	3.7	Mostly Lint; Few clumps Agglomerated dust particles.
-208 +177	4.2	4.7	Lint; Dust, Agglomerated particles.
-177 +147	3.6	4.0	Lint, Agglomerated dust particles.
-147 +61	9.2	10.2	Lint, Dust particles, Agglomerated particles.
-61 +46	9.6	10.6	Lint, Dust particles, Agglomerated particles.
-46 +43	2.5	2.8	Lint, Dust particles, Agglomerated particles.
-43	48.8	54.2	Fine Dust Particles.

concern with which I am connected is a major manufacturer of viscous impingement panel type filters. We do not accept nor publish such figures as representative of the efficiency of the panel type filters which we manufacture. Also, I do not know of any method having the status of a recognized code which will yield such results on a good panel type filter. The authors should be more specific in describing the filters and the code to which they are referring.

In the discussion under the heading Efficiency Versus Particle Size, the authors make the statement, *Hence, a downstream electro-static filter often picks up more dust than the upstream filter.* In inspecting the efficiency results described under the heading Evaluation of Self Charging Electro-Static Air Cleaners we note that the data plotted in Fig. 11 show an average efficiency of 67 percent and those plotted in Fig. 12 show

an average efficiency of 70 percent. Those plotted in Fig. 13 show an average efficiency of 85 percent. If two of these filters are used in tandem, it is obvious that #2 filter could not pick up more dust than #1 filter if the efficiency of the #1 filter exceeded 50 percent. Consequently, if the #2 filter often picks up more dust than the #1 filter, it can only be concluded that the test results often do not apply to a filter under its working conditions. In this respect, the test method displays the same defect for which other proposed methods have been severely criticized.

It should be further noted that test results by other workers in the field do not confirm the effectiveness of panel filters, working on the "self charging electrostatic" principle off either fine or coarse<sup>10, 11</sup> dust.

In examining Fig. 15, it is our understanding that the plot represents the progressive magnitude of the airborne dust load in a closed and sealed room where the air is being re-circulated through a filter. If this is correct, I do not understand why the air gets cleaner for the first 10 hr of running and after reaching a maximum state of cleanliness, it gets dirtier again until it reaches about the same stage of cleanliness that it had after 4 hr of running.

M. E. JACOBY,\* Newark, Ohio, (WRITTEN): We do not agree with the authors that the test they propose is suitable for evaluating panel-type air cleaners. Our experience with *atmospheric dust* indicates that it is highly unreliable for testing, and that many precautions must be taken before significant conclusions can be drawn from such data.

In the absence of precautions necessary for a valid comparison, supporting data should be given. For example, the dust concentration in the test room is compared with a discoloration measurement obtained before starting the test. To interpret the figures, much information is needed about this preliminary measurement:

1. Was the preliminary *comparison spot* based on one measurement or several? Were they as widely variable as the points which appear in Figs. 10-14?
2. What was the dust concentration at the start of each test?
3. What were the conditions of temperature, humidity, etc., under which the tests were run? What were outdoor dust and wind conditions? How long was the room closed before the test started?

As another example, it is stated that *in initial tests there was some air leakage into the room; however, this was later eliminated*. This is obviously an over-simplification—for one thing, 43.6 cu ft per hour were exhausted through the sampling pump. The question naturally arises as to how much leakage really occurred during each of the tests. Was it affected by outdoor wind velocities? Would this help to explain the wide fluctuations in the dust level of the room during the test?

There are other cases where questions arise; the point is, however, that much supplementary data is required for proper evaluation. Since such data was not presented, the conclusions about so-called self-charging electrostatic filters seem a little reckless.

We feel the authors' use of the word *efficiency* should be corrected. In Figs. 10-14 the ordinates should more properly be labeled relative dust concentration vs. running time. We are not sure of what the dust concentration is relative to, but after about an hour's running time it does not seem to decrease any more. From the authors' description, we conclude that no dust was being caught. By all past definitions the filter efficiencies would have been called zero.

L. L. DOLLINGER, JR., Rochester, N. Y., (WRITTEN): In order to hasten to our comments on the test method and results, we concede most of the preliminary discussion of the difference between atmospheric and artificial dusts, the desirability in some cases of testing filters under operating conditions and the efficiency of the *single pass* type of electrostatic filter as demonstrated so aptly in Figs. 7 and 8.

<sup>10</sup> New Developments in Air Cleaning, by Leslie Silverman. (*American Industrial Hygiene Association Quarterly*, 15:3, September 1954).

<sup>11</sup> Discussion, by C. B. Rowe (ASHVE TRANSACTIONS, Vol. 58, 1952, pp. 66, 74).

\* Owens-Corning Fiberglas Corp.

We do not immediately comprehend why, after a prolonged discussion of the desirability of testing filters with outside atmospheric dust, the authors proceeded to test their filters in a room which was carefully sealed to prevent the infiltration of any outside air and airborne dust.

As we understand it, a dust spot was taken from the air in the test room at the beginning of the test and all subsequent readings were compared with the first reading. Therefore, the values which were plotted in Figs. 10 to 13 as efficiencies are actually percent cleanliness of the room air as compared with the original reading or, reading in reversed direction, one gets percentage dirtiness of the air as compared with its original condition. Without, at the moment, discussing the desirability of sketching an overall average for points which vary as much as they do in these graphs, the significance of the straight line average is that the filter did not have an efficiency of 10 or 87 percent as designated, but rather an efficiency of zero from the time at which the average efficiency was reached.

We question the significance of the overall average as indicated by a straight line because it would seem that the user of the filter would be interested in knowing the variation in efficiency from hour to hour, since the greatest smudging of protected goods or equipment would take place at points of minimum filter efficiency. We further question the significance of curves drawn through points representing average efficiencies at given running times. In some cases there is as much as a two-fold variation in the efficiencies at the same running time for three separate series of tests. It might have been more instructive to plot the curve for each series of tests separately but it is not possible to do this in the absence of published data or some means of distinguishing between points for the separate runs on the graphs. Therefore, for the sake of discussion, let us consider the graphs as drawn. Taking Fig. 12 as an example, the dirtiness of the test room is increased from 20 to 50 percent of its initial dirtiness between the seventh and the twelfth hours of running time. If one leaves out of consideration variables other than the efficiency of the filter, this means that the filter had a negative efficiency to the extent that enough dirt was blown out of the filter back into the room during this 5-hr interval to increase the dirtiness of the air to 250 percent of its dirtiness at the beginning of this 5-hr period. Wherever a curve has a negative slope, this means a negative efficiency. It would seem that it might be possible to find some kind of function involving the derivatives of these curves which would be more representative of the actual hour-by-hour efficiencies.

Mention was made of the fact that, after the beginning of the tests, leaks were discovered in the seals about the door and windows. If the readings taken during the time of leakage are represented by the horizontal row of points across the time axis in Fig. 10, they should, of course, have been omitted. The authors are not clear as to just what readings were taken under non-standard conditions.

Whereas the viscous panel exceeded its average efficiency during the first sampling period by more than 100 percent, the filter represented in Fig. 13 did not attain to its average until after more than 3 hr, corresponding to about 60 passes of the same air through the filter. A rather large number of passes was also required in the case of the filters represented in Figs. 11 and 12.

In view of the wideness of the scattering of experimental points on the graphs, we are inclined to think that all the scattering is not due to the characteristics of the filter but that probably a large portion of it finds its origin in the test method. We feel that there must have been considerable sedimentation of dust within the room during the test period and there might well have been a cycle in this sedimentation caused by a repetition of conditions as dust was deposited first in one crevice or on one protected surface and then picked up and deposited elsewhere. The cyclic nature of the conditions is indicated by the repetition of minimum at the end of 12 hr and by the striking resemblance between the various curves.

In their introductory paragraph, the authors discussed the fact that the carbonaceous matter in atmospheric dust is the portion which causes the greatest soiling or, in other words, is removed from air most readily by thermal or electrical precipitation or ordinary

sedimentation. It is also well-known, of course, that soot adheres readily to most surfaces. These facts together with the observation that the percent cleanliness of the air was greater after soot was introduced seems to corroborate our previously-mentioned suspicion that this method of testing involves too many variables due to the possibility of sedimentation of dust in the test chamber.

One incidental question which comes to mind is how to account for the finite efficiency at zero running time represented by the graphs in Figs. 10, 11 and 12. Perhaps this was an oversight in plotting.

In view of the foregoing comments, we can summarize by saying that we question the validity of this method of testing filters because (1) while the filters are presumably tested with atmospheric dust, the tests are made in a room which is sealed against the infiltration of atmospheric dust, and (2) because we feel that the test conditions are such as to give variable results entirely independent of the characteristics of the filter. Even if the test method were valid, we feel that it would not be of general interest but only of interest to those who are interested in filters through which it is necessary to pass the air as many as 60 times before achieving the desired results.

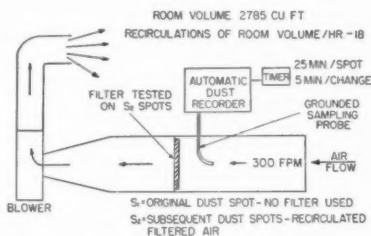


FIG. G. TESTING ARRANGEMENT

**AUTHORS' CLOSURE:** During the presentation of the paper, the authors introduced an additional illustration, Fig. G, which illustrates the testing arrangement. It should be noted that no external discharge for the sampling pump is shown as indicated in Fig. A of the discussion by Professor Algren and Dr. Whitby. The air from the sampling pump was discharged into the room. Otherwise Fig. G is self-explanatory.

Professor Algren and Dr. Whitby contend that the authors' statement that the problem of air cleaning is largely one of removing finely divided carbonaceous matter, is an over simplification of the air cleaning problem and that our quoted figure of 65 percent carbonaceous matter in atmospheric dust is high as indicated by their figure of 35 percent supported by only five samples. It is relatively unimportant whether the figure is 65 or 35 percent, but we believe most filter manufacturers agree it is important to remove carbonaceous matter from the air. Carbonaceous matter is the source of the damaging effects on walls, woodwork, drapes and merchandise, whereas, the other contaminants may be wiped off without damage.

Professor Algren and Dr. Whitby also point out that a considerable portion of the free carbon in atmospheric dust adheres to the larger dust particles under normal conditions. This is not substantiated by the performance of impingement type filters or by microscopic examination of cascade impinger test slides.

Referring to the work of Dalla Valle, Orr, and Hinkle, the statement is made in the discussion of Professor Algren and Dr. Whitby that normal electrical charges on the particles affect the shape of aggregates of particles but not the aggregation rate. We do not believe that this statement is applicable in the tests cited because we have shown visual proof that the agglomerate size is increased as a result of passing through a self-charging electrostatic filter as exemplified in Fig. 6. We know the original particle

size of the carbonaceous matter was  $1\text{ }\mu$  or less, and after passing through the filter the particles became  $150\text{--}250\text{ }\mu$  in size.

The particles shown in Fig. 6 were taken from the Army Base Telephone Exchange in Oakland, California, 30 ft down stream from a self-charging electrostatic filter. These were not obtained from the enriched air stream and they are normal atmospheric dust particles. We have obtained the same type particles down stream in several other localities. We have seen numerous instances where this has occurred. Without the electrostatic air cleaner in the air stream they would be invisible.

We are glad to learn that the wide fluctuations clearly visible in Fig. 1 are recognized and we agree that they probably have a bearing on the fluctuations in Fig. 10, however, careful examination of Figs. 11 to 14 does not reflect the wide fluctuations.

We do not agree that the rate of infiltration of air into the test room is high enough that this factor rather than the filter efficiency is the most important factor in determining the values shown in Figs. 10 to 14. Fig. H shows the results of 542 tests as shown in the curve. We used an *ATSI* recorder inside the room, passing a column of air 5500 ft in length through No. 4 filter paper at a velocity through the filter paper of 46 fpm. An identical recorder was used to measure the dust contained in the outside air so that simultaneous dust spots could be matched. Atmospheric dust content in terms of percent reduction of transmitted light is plotted against the filter efficiency as determined by direct comparison of matched inside and outside spots.

In order to make the tests severe and rigorous, the door to the test room was opened 174 times per day so that the infiltration of air in the room was abnormally high. Here, the overall cleanliness of the air in the room compared to the outside atmosphere, which we have termed filter efficiency, is 48.39 percent. The authors believe that this value represents abnormally high infiltration of air and compares directly with the results of Fig. 12 where the average efficiency is 70.08 percent with little air leakage. We disagree that under our test conditions any filter can theoretically give 100 percent efficiency, if allowed to run long enough. However, in each one of these tests cited there were 432 volume changes of room air passing through the test filters which should be ample time to establish the practical filter performance. Also referring to Figs. 10 to 14 we believe that the trend is clearly established that no further improvement of filter performance will be achieved by continuing the test.

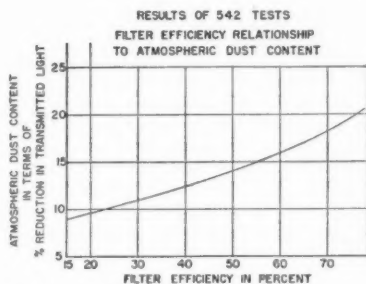
In Table B of the comments by Professor Algren and Dr. Whitby we come to the value of  $L$  representing air infiltration in cubic feet per minute. We note 60 cfm is the air infiltration assigned Fig. 10. It seems unreasonable that this value, which should be a constant, is 3 to 6 times higher than for the tests exemplified in Figs. 11, 12, 13 and 14. When we compare air infiltration values for Figs. 11 to 14 we find

FIG.	CFM	PERCENT
11	11.2	1.35 of 833 cfm
12	19.3	2.3
13	10	0.8
14	12.4	1.5

Therefore, air infiltration plays only a small role in these tests.

Furthermore, we disagree with one assumption in the equation of Rowley and Jordan. The dust concentration produced inside per minute ( $D$ ) does not equal zero and, therefore, casts doubt on the validity of values obtained in Tables A and B. Dust was being produced constantly by the V-belt drives on the tunnel and sampling pump. It was also known, as stated in the authors' paper, that the oil bath silencer on the sampling pump was emitting oil droplets into the air.

When the equation is simplified, the term  $D$  is present. Therefore, this value has played a major part in causing fluctuations in the efficiency curve not only the term  $L$ .



(Note: Test room door opened 174 times per day July 26, 1954-August 24, 1954)

FIG. H. CURVES SHOWING RELATION BETWEEN FILTER EFFICIENCY AND ATMOSPHERIC DUST COUNT RESULTING FROM 542 TESTS

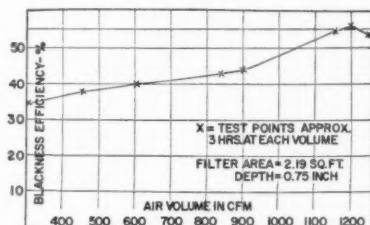


FIG. J. CURVE SHOWING SINGLE PASS BLACKNESS EFFICIENCIES AT VARYING AIR VOLUMES

Therefore, air leakage cannot be blamed entirely for the variations in Fig. 10. It is the author's opinion that the variations shown in Fig. 10 well represent the difficulties encountered in testing ordinary viscous impingement type air cleaners on atmospheric dust.

Statement is made that it is doubtful if the single pass efficiency of any filter tested is above 15 percent. This apparently is a matter of opinion and we accept the opinion of Professor Algren and Dr. Whitby on this value. In fact, the integrated efficiency on the curve in Fig. H indicates that with low dust content in the air it is possible to achieve 15 percent, which is lower than single pass efficiency. However, since attention has been directed to single pass efficiency and values quoted, we would like to introduce Fig. J, a curve showing single pass efficiencies run on self-charging electrostatic filters at varying velocities where each test is a result of 3 hr testing on matched spots and the single pass efficiencies range from 34 to 55 percent using the National Bureau of Standards Test Method on atmospheric dust.

The authors should like to emphasize that their paper does not deal with single pass efficiencies but with the integrated efficiencies as represented by room air cleanliness.

Comment has been made on the 533 fpm velocities through the filter paper of our sampler. The authors recognize that this velocity is high and here again would like to point out that on the curve, Fig. H, the velocity through the filter paper was 46 fpm and the results are substantially in agreement with the 533 fpm. We feel this is proof that the velocity through the filter paper does not have much bearing on the results.

The use of high efficiency filter papers is desirable but impractical due to the large number of tests necessary.

We value Professor Algren's and Dr. Whitby's comments, however, we feel air cleaning efforts should be directed toward the measurement of the inside air cleanliness just as the *Air Pollution Control Association* directs its efforts to accurately measuring outside air dirtiness.

As regards the discussion by Professor Silverman and Mr. Anderson the authors would first like to state that they do not claim that the use of atmospheric dust for testing filters is new.

Carbonaceous dust is responsible for most of the soiling which makes house cleaning and redecorating necessary. Other dusts may be more important [from a public health standpoint, but these dusts were not the subject of our paper.

Accelerated filter evaluation tests with special dusts are like accelerated weathering

tests on paints. They give an answer, but correlation with actual atmospheric tests is questionable. Twenty-four hours is not considered too long a time for an accurate evaluation of filter efficiency. Dust holding capacity is of secondary importance and is not the subject of the present paper.

We admit that our test procedure is a deviation from *standard practice*. This was intentional as the previous methods have been found wanting by many investigators and the authors consider that a new approach is necessary and desirable.

It is recognized that the method of testing with smoke enriched atmospheric dust needs improvement and refinement. In a previous paper<sup>12</sup> the authors described a method of controlling the smoke concentration and its characteristics. It was found that carbon black or lamp black cannot be dispersed with an ejector and that the carbon must be generated in situ.

Professor Silverman should check reference 9, on the Reference List of the authors' paper, (p. 69), on the use of cyclones and bags before electrical precipitators. The rubber industry would certainly like to know if the manufacturers of furnace blacks are using the wrong sequence of equipment in their process.

The statement concerning 1 to 2 air changes per hour does not agree with p. 142, line 8, of THE GUIDE, 1954—A ventilation rate between 8 and 12 changes per hour is desirable—. In the average home the recirculation may run from 120 to 240 changes per day.

The drop in Fig. 10 to low values is covered in our comments on Mr. Dollinger's discussion. The room air volume drawn through the sampler is not a factor in the filter performance because it applies equally in all cases. The discharge from the sampler was not exhausted from the room and, therefore, cannot be classified as needing replacement.

The authors disagree with the statement that, if over the period of several hours, an impingement filter is operating at 20 percent efficiency it will have removed practically all of the dust in the room. Fig. 10 shows no indication of removing all dust, only 11 percent of the total. Fig. 5 shows clearly what happens to fine particles. They go right through and few are caught even by recirculation. Hence there are fine particles which never will be captured by viscous impingement filters.

In reference to the filter paper used, we chose the No. 4 paper on the basis of all the data presented by Smith and Suprenant. In their Table III (reproduced here) the particle count was reduced more by the No. 4 paper except for the 0.6–0.8  $\mu$  fraction. In their Table IV (also reproduced) are the values referred to by Professor Silverman:

COUNT ANALYSIS OF AIRBORNE PARTICLES—PERCENT OF EACH SIZE (TABLE III)

PARTICLE DIAMETER MICRONS	UNFILTERED AIR	FILTERED AIR	
		PAPER No. 4	PAPER No. 41
below 0.4	31.8	45.2	47.4
0.4–0.6	42.6	38.1	40.3
0.6–0.8	16.2	14.2	9.1
0.8–1.0	6.8	1.9	2.4
1.0–2.0	1.6	0.6	0.8
over 2.0	1.0	...	...

Prof. Silverman seems to believe that the efficiency of the No. 41 paper is better because of his quoted values of 64 percent and 38 percent in the 0.6  $\mu$  to 0.8  $\mu$  range. But Table IV indicates that the 0.6  $\mu$ –0.8  $\mu$  range is the only one where No. 41 outperforms paper No. 4.

<sup>12</sup> Self-Charging Electrostatic Air Filters, by W. T. Van Orman and H. A. Endres (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 53).

It is immaterial which paper is the better because the effects of the sampling are cancelled out when comparing any of the curves. Also air pollution recorders as developed by the American Iron and Steel Institute and other recorders use No. 4 paper.

The efforts to show that reentrainment influences the tests is most determined. There is no evidence of this in Fig. 10 and the high level of performance shown in Figs. 11 through 14 refutes this contention.

The reason the room does not become 100 percent clean is simple. No filter is 100 percent efficient and with this system we reach a point where the particles are of such size, shape and density as to preclude further cleaning.

#### FILTERING EFFICIENCY BY PARTICLE SIZE FOR EACH OF SEVERAL AIR SAMPLING MEDIA (TABLE IV)

PARTICLE DIAMETER MICRONS	FLOW RATE OF 20 FPM	
	PAPER No. 4	PAPER No. 41
below 0.4	23%	23%
0.4-0.6	22	28
0.6-0.8	38	64
0.8-1.0	79	74
1.0-2.0	74	70
above 2.0	100	100

Regarding single pass efficiencies see Fig. J and our comments covering the discussion by Professor Algren and Dr. Whitby.

Anyone interested in the nature, magnitude, persistence and measurement of electrostatic charges on high dielectric plastics is referred to the work of Woodland and Ziegler reported in *Modern Plastics*, May 1951, p. 102. In this paper it is pointed out that electrostatic charges on plastics can only be measured by specialized techniques. This is probably the reason why Professors Silverman and Anderson have found no evidence of any charge.

Fig. H shows clearly that with increased dust content improved efficiencies are obtained. We know from National Advisory Committee for Aeronautics tests that dust is necessary to produce electrostatic charge. Six years' experience shows the self-charging electrostatic filter has adequate holding capacity.

Reentrainment is not a factor in these tests. The use of the height of air column and square feet of standard discoloration was suggested by an outstanding authority on air cleaning and offers a new perspective on a complex problem. The authors believe that this perspective has value and should not be rejected lightly by mere statement of opinion.

In conclusion, we wish to point out again that the objective of the paper was testing panel type air cleaners by means of atmospheric dust, and was not concerned with the type of filters.

The authors agree with Mr. Farr that the use of a uniform standardized test dust for the evaluation of filter performance would be highly desirable, but such a dust must be more nearly representative of average normal atmospheric dust than any thus far devised.

The fine particles present in atmospheric dust are capable of remaining suspended in the air for long periods of time, but they do constitute a soiling problem due to electrostatic and thermal precipitation. Hence the soilage of walls and ceilings.

Mr. Farr's point on the possibility of careless interpretation of Figs. 10 through 14 is well taken and the authors hope the reader will take sufficient time and effort to clearly understand them. Recirculated air is being widely used in homes, offices, and industry today and, therefore, may properly be considered part of the problem.

The authors feel that they should correct Mr. Nutting on the number of air changes per hour. There were actually 18 air changes per hour.

Mr. Nutting's choice of the 2-hr point in Figs. 11 through 14 is seriously open to question. The air cleaning processes listed 1 through 4 apply equally well to viscous impingement filters so they too are being given equal credit for work which they did not do in cleaning air with the result that all averages are somewhat higher. Let us choose the 10-hr period for the discussion of Fig. 11. Here the performance is close to zero. The selection of a single point in Fig. 11 enables anyone to find what he wants to find, but here again the emphasis should be directed toward average values. In the author's 1952 paper (p. 66) we gave the Reference 2 Static Dust Collection on Plastics, by P. C. Woodland and E. E. Ziegler. In this reference on p. 15 we find this statement.

This work has also shown that electrometer readings are valuable only when obtained and interpreted by a person completely familiar with the use of the instrument. In this respect the situation resembles that of cross-polaroids which can be used constructively to indicate certain conditions within a plastic molding, but which can be misused by the novice to indicate conditions that do not exist.

Thus it is that a *neutral* condition on a molding might be indicated by an electrometer and might well be interpreted to mean that no dust will collect on that molding. The fact might be that a very high positive and negative charge, existing side by side, are balancing each other out as far as the instrument is concerned, and that unusual and heavy dust deposits might be expected in both areas.

The authors present the following answers to the three questions mentioned by Mr. Nutting.

1. The great distances smoke travels disputes the piggy back theory. Also under Mr. Farr's comments he pointed out he could find no particles under  $10\ \mu$  in his sedimentation test plates.

2. The electrostatic and thermal precipitating power of the walls is insignificant compared to that of the electrostatic filter. This is shown by the cleanliness of walls and ceilings of rooms in buildings where electrostatic filters are used.

3. Fig. J gives the answer obtained by the National Bureau of Standards Method. The ASHAE filter test is now considered obsolete. No formal action of either approval or disapproval has been taken by ASHAE on the AFI test code.

The authors agree with Mr. Landgraf that sedimentation is a factor in this system without question. It was our purpose in introducing a control viscous impingement filter to provide a comparative level of filter performance. If we knew how to eliminate sedimentation it would reduce the levels of Figs. 10 to 14 by an equal amount, say 5 percent.

The authors appreciate Mr. Evans' comments.

In reply to the comment by Dr. Penney the authors would state that the inclusion of smoke enriched atmospheric dust tests was prompted to provide a means of testing for emergency use under conditions of unusually clean air which make testing impossible otherwise. We heartily agree the actual atmospheric dust is the best way to test air cleaning devices. Fig. H shows the long time test of sufficient number to answer Dr. Penney's thoughts on short time efficiencies. The authors recognize Dr. Penney's service in the air cleaning field and express their appreciation of his valued comments.

The authors are pleased with the agreement of Mr. Gregory to the test values for viscous impingement filters shown in Fig. 10. One of the country's outstanding authorities has said that improvement in air filtration must come in the better collection of the finer particles. Our paper suggests a test method which is a practical application of evaluation on the real problem—removing atmospheric dust. The authors have commented on the AFI test code many times through the *Open for Discussion* columns of *Heating, Piping & Air Conditioning*. We are not in favor of the code since it does not represent atmospheric dust.

Mr. Avery points out the need for larger fan motors to permit continuous fan operation, with which we heartily agree.

Mr. Hemeon points to the need of correctly matched photocell and ammeter. These were calibrated to insure that the electrical output of the photocell was directly proportional to the incident light.

Fig. G presented at the meeting should clarify the foundation of the mathematics used. All spots were compared to the original dust spot  $S_1$ .

Fig. 13 represents the only conditions under which the atmosphere was enriched by artificial smoke.

Every effort was made to prevent infiltration of outdoor air by sealing all crevices with masking tape to minimize the influence of diurnal and other cyclic changes of atmospheric dust content. We believe the infiltration influence has been held to a sufficiently low value so as not to appreciably affect the results of our tests. The hills and valleys of our curves may also be influenced by the discharge of the sampling pump and the v belt drive to the blower.

We believe Figs. 3, 4 and 5 illustrate Mr. Hemeon's thoughts that there is a sharp dividing line between fine and coarse particles in their behavior and properties. We suggest that  $10\ \mu$  may well be considered as the boundary line. It happens to be the dividing point between visible and invisible particles which seems to add stature to this boundary. Classification of dust particles from centrifugal separators seems to fit very nicely this  $10\ \mu$  dividing line. The authors congratulate Mr. Hemeon for the work he has done in the *Air Pollution Control Association* and fervently hope that his suggestion that we have a fine and coarse dust problem to be treated separately may find acceptance in the Society. We believe the fine dust problem has been neglected much too long.

It is the privilege of Mr. Jacoby to disagree with our tests, but when he indicates atmospheric dust is *highly unreliable* for testing, we believe the word should be *variable*. Atmospheric dust is the condition encountered and its removal, the objective of air filtration. The Wright Brothers were not daunted by variable wind velocities or is modern aviation completely stopped by variations of weather. Testing on atmospheric dust is not easy, but it is the problem and should be faced.

All the precautions of a research laboratory were used to guard against the many variables involved materially influencing the results. Representative dust spots were submitted but space and printing accuracy precluded their inclusion. To answer the questions listed by Mr. Jacoby:

1. The preliminary dust spot was not based on one spot and the spots were not widely variable.

2. The dust concentration at the start of each test was 25 percent reduction in transmitted light for a column of air 13,400 ft in length or a greater concentration.

3. The temperatures were 77 F, humidity was not controlled. The outdoor dust and wind conditions were isolated by sealing the room. The room was open 24 hr before each test run and the door closed and sealed immediately after the  $S_1$  spot.

The 43.6 cu ft per hour discharge from the sampling pump was discharged into the sealed room and, therefore, may not be considered as being replaced by leakage.

The supplementary data exemplified on Fig. H should do much to satisfy the requirements of this method of testing. This was covered in our answer to the comments of Professor Algren and Dr. Whitby.

Figs. 10-14 represent the effectiveness in cleaning the air and, therefore, may properly be called efficiency. It should not be confused with single pass efficiencies. Seventeen times in the text we have mentioned recirculated air and we believe no one can misunderstand our intentions.

With reference to the remarks of Mr. Rowe, the authors believe, as Mr. Hemeon has pointed out, that the problem of cleaning air is a two-step problem, one of fine particles and one of coarse particles. Our paper covers atmospheric dust and the measuring technique emphasizes fine particles. We note in Table C and Fig. F of Mr. Rowe's discussion that under  $43\ \mu$  are classified as fine particles. We submit that  $10\ \mu$  and less is a better classification. The sealed room is an excellent research approach enabling a control of variables not possible before. Fig. H shows what happens

in an occupied room. Mr. Rowe questions the value of 11.05 percent being in line with viscous impingement tests. We can cite at least one case. The authors recall that in a booth of a filter manufacturer in the 1953 ASHAE show there was a display of filter efficiencies. Using the Bureau of Standard Test Method that manufacturer showed 5 percent efficiency for various impingement filters.

Regarding Mr. Rowe's comments on the paper which two of the authors presented in 1952, we refer to the TRANSACTIONS ASHVE, Vol. 58, 1952, pp. 75 and 76.

Regarding the comments of Mr. Dollinger, the authors point out that in their test method they allow the test chamber to become full of atmospheric dust in normal concentration. Then the chamber is sealed off to prevent atmospheric variations from affecting the results. By this step they can control the variables more closely. This is simply good research procedure.

The average efficiency as shown in Figs. 10 through 14 is the index of performance of the filter from the start to finish of the test. The point to point efficiency after the average line is reached is zero, but this average line serves as a yardstick in evaluating the overall performance of the filter.

Hourly variations in efficiency are of little use to anyone. In choosing a filter for your own use would you choose a filter with an overall average of 10 percent or one with 70 percent? Hourly variations mean little by themselves. Also it is not a matter of maximum smudging power at minimum efficiency, for it happens during the summer months that the efficiency is close to zero yet the atmosphere is exceptionally clean and has less smudging power than at other times—Fig. H exemplifies this fact.

The negative efficiencies referred to in Fig. 12 are not due to the filter performance alone. Therefore, any comment involving only a segment of the curve is as useless as taking a word out of text. Fig. H should clarify any hourly variations.

Leakage was discovered during the initial test runs; however, no such data is included in the paper.

The time involved in reaching the overall average is of little significance. The curves were meant to represent how well the air has been cleaned. The overall averages speak for themselves.

The so-called wide scattering of points in our curves as mentioned by Mr. Dollinger is caused by several factors. Because the room was sealed off, it was difficult to control temperature, pressure, and relative humidity. By increasing the number of filters as shown in Fig. 14 and in further tests with 3, 4 and 5 filters in depth the points were not widely scattered. It has been our experience that panel type filters exhibit cyclic performance, and all this cannot be blamed on the method.

In referring to sedimentation Mr. Dollinger misquoted our words. The term the authors used was *sedimentation after flocculation*, not *ordinary sedimentation*. The particles of the soot were too small to allow them to settle out. Agglomeration occurred first so the particle became large enough to be captured.

Concluding our comments, we like the truly automatic method of recording dust concentrations and credit should be given to the *Air Pollution Control Association* for making this development possible. That group tells us how dirty the air is outside and by applying the same method we can measure the cleanliness of air of spaces intended for human occupancy.

It is our hope that the reader will not believe we have provided all the answers to the problem but we do feel gratified by the interest and number of comments received from so many noted authorities in the air cleaning field.



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## PRELIMINARY STUDIES OF HEAT REMOVAL BY A COOLED CEILING PANEL

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**This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS, INC., at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.**

**H** EAT EQUIVALENT to one-half ton of refrigeration (6000 Btu per hr) can be removed from a living room of normal size by a cooled ceiling panel alone, operating only 10 deg below room temperature. This fact has been demonstrated by experimentation in the ASHAE Environment Laboratory under the research program planned by the TAC on Panel Heating and Cooling.

The program will supply reliable design data for panel-cooling systems. Such systems are becoming increasingly popular. Although this is not the final report on the subject, it does give information on factors that are important in design and those which are not.

Following the completion of the studies of panel-heating, an investigation of space cooling by cooled ceiling panels was started. The complete panel-cooling program, as outlined by Group B‡ of the TAC on Panel Heating and Cooling, includes four phases. The first, which is the subject of this paper, is concerned with the heat exchanges that take place in a ceiling panel-cooled space under various conditions of surface temperatures, infiltration and furnishings; but does not include the effects of the introduction of conditioned air, luminaires, or solar energy. In the second phase, now in progress, heat-transfer relationships in a room cooled by a combination of a cooled ceiling panel and conditioned air are being studied. The third and fourth phases will include the modifying effects of luminaires and solar radiation on panel performance.

The final report on the panel-cooling program will probably be presented soon after mid-year 1955.

### CONCLUSIONS

1. Ceiling-panel cooling is an inversion of floor-panel heating, and the performance of a cool ceiling panel can be predicted from the performance of a warm floor panel.

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TABLE 1—ROOM AIR TEMPERATURES AND HEAT REMOVAL BY COOLED CEILING PANEL  
(24½ x 12 x 8 FT High Room)

GENERAL TEST DESCRIPTION	TEST No.	ROOM SURFACE TEMPERATURE, F							AUST <sup>a</sup>	INFILTRATION		HEAT REMOVAL BY PANEL Btu/(hr) (sq ft)		ROOM AIR TEMPERATURE, F				
		Ceiling	Floor	North Wall	South Wall	East Wall	West Wall	Changes per hr		Temp. F	Observ.	Calculated <sup>b</sup>	Observed		Calculated <sup>b</sup>			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17		
Uniform Environments	281	65.6	92.6	89.9	92.0	92.8	92.5	92.2	None	—	39.0	38.0	83.7	83.9	83.7	83.7	83.6	
	282	64.7	84.9	83.9	85.6	85.1	85.6	85.1	None	—	28.5	28.0	79.0	78.6	78.6	78.6	78.5	
	363	59.5	85.3	82.7	85.2	85.3	85.1	85.0	None	—	35.0	35.5	77.8	77.2	76.8	76.8	76.8	
	364	68.9	86.6	84.8	86.7	86.8	86.7	86.5	None	—	24.5	24.5	81.4	81.3	80.9	80.9	80.8	
	365	78.9	84.7	84.3	84.9	85.8	84.9	85.0	None	—	28.0	8.0	83.2	83.2	83.0	83.0	83.0	
	366	69.2	84.8	83.9	85.7	86.2	85.7	85.3	None	—	22.0	22.5	80.8	80.3	80.1	80.1	80.1	
	367	69.4	84.8	83.3	85.0	86.0	85.0	85.0	None	—	21.6	21.5	80.5	80.5	80.0	80.0	80.0	
	368	59.3	70.0	69.4	70.0	70.2	70.0	70.0	None	—	13.0	14.0	66.7	66.6	66.6	66.6	66.6	
	369	60.2	79.8	77.9	79.8	80.1	79.8	79.7	None	—	25.5	26.0	73.5	73.3	73.5	73.5	73.4	
	370	58.5	105.2	102.0	104.7	105.2	104.6	104.7	None	—	69.0	72.5	91.2	90.6	89.9	89.8	89.8	
	371	69.3	104.7	100.7	104.7	105.1	104.5	104.3	None	—	55.0	53.5	94.3	93.3	93.3	93.1	93.0	
	372	69.8	95.0	92.3	95.2	95.4	95.1	94.9	None	—	36.0	37.0	87.4	87.0	86.9	86.8	86.8	
283	64.9	86.0	83.9	85.7	84.8	85.6	85.4	1.0	110.2	30.5	30.5	80.7	80.3	80.2	80.2	80.1		
391	70.0	84.1	83.7	84.2	84.9	84.1	84.2	1.05	110.0	21.0	21.0	80.7	80.6	81.0	80.9	80.9		
392	70.3	85.2	84.2	84.9	84.1	84.7	84.7	1.9	110.2	23.0	23.0	82.5	82.6	82.9	82.6	82.6		
393	70.7	85.2	84.5	85.2	85.2	85.0	85.1	2.9	110.2	24.0	23.5	84.0	83.8	84.6	84.2	84.2		
387	69.8	84.4	83.9	84.7	84.6	84.6	84.5	1.0	79.6	19.5	20.5	79.7	79.7	79.3	79.3	79.3		
388	69.8	84.2	83.9	84.7	84.6	84.7	84.4	1.9	80.2	20.5	20.0	79.8	79.5	78.8	78.8	78.9		
389	70.2	84.7	84.1	84.8	84.8	84.8	84.7	2.9	80.2	19.0	20.0	80.1	79.7	78.5	78.5	78.8		

TABLE 1 (CONTINUED)—ROOM AIR TEMPERATURES AND HEAT REMOVAL BY COOLED CEILING PANEL

Non-Uniform Environments	394	70.0	85.1	84.8	84.9	108.8	85.0	90.3	None	—	29.0	28.5	84.3	84.3	83.8
One long wall hot, all others 85 F	395	70.2	85.2	84.9	85.1	108.6	85.0	90.3	1.0	110.0	30.5	29.5	85.4	85.4	84.9
One long wall hot, 3 walls & floor neutral	396	70.5	90.6 <sup>d</sup>	101.2 <sup>d</sup>	100.7 <sup>d</sup>	109.2	101.0 <sup>d</sup>	99.3	None	—	41.0	42.5	90.6	90.6	90.0
	397	69.9	79.0 <sup>d</sup>	82.6 <sup>d</sup>	82.0 <sup>d</sup>	85.0	82.1 <sup>d</sup>	81.7	None	—	15.0	16.0	78.0	78.4	77.9
3 simulated windows 110 F, 2 simulated outside walls, floor and 2 walls neutral	398	69.6	80.2 <sup>d</sup>	80.2 <sup>d</sup>	89.6 <sup>d</sup>	89.5	80.6 <sup>d</sup>	83.4	None	—	19.0	19.0	79.0	79.0	79.0
	399	64.5	77.8 <sup>d</sup>	77.6 <sup>d</sup>	89.5 <sup>d</sup>	89.6	78.3 <sup>d</sup>	81.8	None	—	23.0	23.5	76.7	76.7	76.2
	400	65.1	81.0 <sup>d</sup>	80.8 <sup>d</sup>	89.5 <sup>d</sup>	89.5	80.8 <sup>d</sup>	83.7	1.0	109.5	27.0	27.5	79.5	79.5	79.0
Room Furnished	401	64.7	77.5 <sup>d</sup>	76.8 <sup>d</sup>	89.3 <sup>d</sup>	89.3	77.4 <sup>d</sup>	81.3	None	—	21.5	23.0	76.6	76.4	77.2
	402	64.4	80.8 <sup>d</sup>	79.6 <sup>d</sup>	89.5 <sup>d</sup>	89.3	79.9 <sup>d</sup>	83.3	1.0	109.3	27.0	27.5	78.8	79.1	78.7
	403	64.8	85.0	83.7	84.8	84.7	84.7	84.7	None	—	25.5	27.5	79.1	78.4	78.3
Internal heat sources	404	60.4	84.8	84.0	83.7	84.6	84.6	84.5	None	—	26.0	—	82.7	82.7	—
One KW	405	69.8	84.8	84.4	84.9	84.8	84.8	84.8	None	—	30.5	—	85.0	85.0	—
Two KW	406	69.9	84.8	85.0	84.9	85.1	84.8	84.9	None	—	37.0	—	87.7	87.7	—

<sup>a</sup> Area weighted average temperature of the floor and walls.

<sup>b</sup> Calculated by the inversion principle for a uniform environment having floor temperature listed in col. 9 and AUST listed in col. 3.

<sup>c</sup> One 4 x 5 ft window in center of south wall. Two 4 x 5 ft windows in east wall.

<sup>d</sup> Neutral surface, neither heated nor cooled by liquid circulation.

2. An appreciable amount of sensible heat can be removed from a room by a cooled ceiling panel, the surface temperature of which is above normal inside dew-point temperatures. In a typical test a ceiling at 10 deg F below room temperature absorbed 20 Btu per (hr) (sq ft) of ceiling area.

3. Infiltration of warm air and internal, convective, heat sources must be considered for design purposes.

4. Non-uniform wall and floor temperatures can be represented by area-weighted average temperatures (AUST)<sup>†</sup> in calculating heat pickup by a cooled ceiling.

5. Room furnishings can be neglected as they affect the heat pickup by only approximately 5 percent.

### TEST EQUIPMENT AND CONDITIONS

Tests were made in the ASHVE Environment Laboratory, which was constructed several years ago for the study of heat transfer within a panel-heated or panel-cooled space. This room is 24.6 ft long, and 12.2 ft wide. For all tests reported in this paper, the adjustable ceiling was at a height of 8 ft. The temperatures of all six surfaces of the room were accurately controlled and measured. Heat flow to or from the surfaces was metered by approximately 150 plate-type heat-flow meters. Heated air was introduced at six locations in the room to simulate infiltration. A detailed description of the construction and instrumentation of the room is given in an earlier paper.<sup>1</sup>

Tests were made with various combinations of panel surface temperature, wall and floor temperatures, and infiltration air quantity and temperature. In some cases, the four walls and the floor were maintained at the same temperature, thus creating a *uniform environment* while in other tests, *non-uniform environments* were created by holding the floor and various parts of the walls at different temperatures. In a few tests, windows were simulated by heating certain 4 x 5 ft panels to a temperature somewhat higher than that of the adjoining wall surfaces. Non-uniform environments were also obtained by neither heating nor cooling a certain area, but permitting it to seek its own equilibrium temperature. Such an area is referred to as a *neutral surface*.

Most tests were made with the room empty; but a few tests were made with room furnishings, and in several others, convection-type heaters were placed in the room.

### TABULATIONS OF OBSERVATIONS AND CALCULATIONS

The results of tests, all of which were made under steady-state conditions, are given in Table 1. A general description of the environment during each test is given in Column 1, and test numbers are given in Column 2. The average surface temperatures of the cooled ceiling panel, and of the floor and each of the four walls, are listed in Columns 3 to 8 inclusive.

The average uncooled-surface temperatures (AUST) listed in Column 9 are the area-weighted average temperatures of the floor and the four walls. The amount and temperature of the infiltration air are given in Columns 10 and 11.

The observed rates of heat transfer to the ceiling panel are listed in Column 12,

<sup>†</sup> AUST signifies Average Uncooled Surface Temperature.

<sup>1</sup> Exponent numerals refer to References.

and the observed air temperatures at the 30-in. and 5-ft levels are given in Columns 14 and 15. For comparison with these observed values, calculated heat transfer rates and temperatures are listed in Columns 13, 16, and 17. These values were calculated from the results previously reported on floor-panel heating<sup>2</sup> by assuming a complete inversion of conditions.

### DISCUSSION OF RESULTS

*Inversion Principle:* Theory suggests that convection heat transfer, as well as radiation exchange, in a room with a cooled ceiling panel, should be the same as in a room with a heated floor panel, if the temperatures of the ceiling panel and the surrounding surfaces of the cooled room are respectively the same as the temperatures of the surrounding surfaces of the heated room and the heated floor panel. Furthermore, the difference in temperature between the cooled ceiling panel and the room air should be the same as between the heated floor panel and the room air.

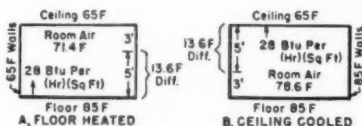


FIG. 1. INVERSION PRINCIPLE

The inversion principle is illustrated in Fig. 1. Room A of Fig. 1 shows a heated floor panel at 85 F, and ceiling and walls at 65 F. For these surface temperatures, previous research<sup>2</sup> has indicated that the room-air temperature at the 5-ft level would be 71.4 F, and the heat output of the floor panel would be 28 Btu per (hr) (sq ft). An inversion of this condition is shown in Room B of Fig. 1. Here the surface temperatures of the cooled ceiling and the walls are 65 and 85 F respectively, and the heat pickup by the cooled ceiling panel would be 28 Btu per (hr) (sq ft). The difference between the temperatures of the heated floor panel and the air in Room A, 5 ft above the panel, is 13.6 F, and this same difference in the temperature between the cooled ceiling panel and the air would indicate an air temperature of 78.6 F in Room B 5 ft below the panel.

The calculated values of heat pickup by the cool-ceiling panel, listed in Column 13 of Table 1, are in good agreement with the observed values shown in Column 12. Similarly, the calculated air temperatures at the 30-in. and 5-ft levels, given in Columns 15 and 17, agree well with the observed temperatures in Columns 14 and 16. This close agreement demonstrated the feasibility of calculating the performance of a cooled ceiling panel from previously determined data on floor-heating panels. It was therefore possible to materially reduce the number of tests which would otherwise have been necessary to completely determine the heat-removal capacity of cooled ceiling panels.

*Air-Temperature Gradients:* Room-air temperature gradients for uniform environment and no infiltration are small, as shown in Fig. 2. Curve A is the gradient which would exist<sup>2</sup> in a floor-panel-heated room with 85 F floor temperature and 65 F AUST. Curve B is the temperature gradient which would be predicted

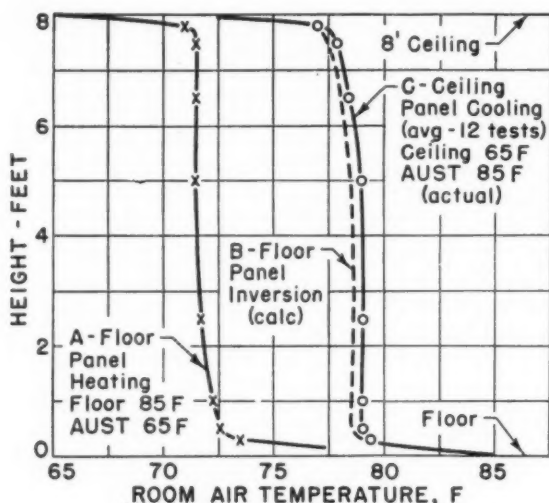


FIG. 2. ROOM AIR TEMPERATURE GRADIENTS FOR COOLED CEILING PANELS AND HEATED FLOOR PANELS; UNIFORM ENVIRONMENT, NO INFILTRATION,  $24\frac{1}{2} \times 12 \times 8$  FT HIGH ROOM

from Curve A by the inversion principle for a ceiling-cooling panel at 65 F and 85 F AUST. The air temperatures indicated at the 4 ft height by Curves A and B are 13.5 F below the heated-floor temperature and 13.5 F above the cooled-ceiling temperature, respectively. The air temperatures observed with a cooled ceiling panel are shown in Curve C. These are only slightly higher than the values shown by Curve B.

**Cooling Performance:** The performance of a full ceiling panel in effectively cooling the 24.6- x 12.2-ft room is shown graphically by Fig. 3. The directrix shows that a ceiling panel at 69 F will remove 21.6 Btu per (hr) (sq ft) from a room which has an air temperature of 80 F. The ceiling area was 300 sq ft and the total heat pick-up was 6480 Btu per hr. All of the heat absorbed by the panel entered the room by conduction through the walls and the floor.

It should be noted that the AUST, not included in Fig. 3, is related directly to the panel temperature, and the room air temperature. The relationship between AUST, panel temperature, and panel pickup was not shown because it would have been the same as Fig. 1 of reference 2 except for the inversion—the AUST of the figure becomes the cooled-panel temperature, and the floor temperature becomes the AUST.

#### FACTORS THAT MUST BE CONSIDERED IN DESIGN

**Effect of Infiltration:** Fig. 4 shows graphically the effects of varying quantities of infiltration air at 110 F, with a fixed ceiling temperature and AUST. Warm

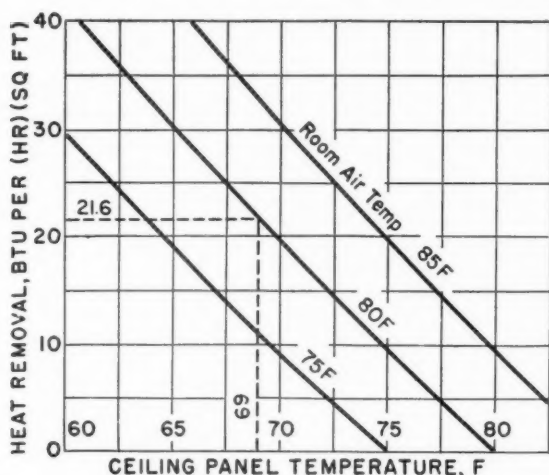


FIG. 3. PERFORMANCE OF A COOLED CEILING PANEL, UNIFORM ENVIRONMENT, NO INFILTRATION, NO INTERNAL HEAT SOURCES

air infiltrating the room caused the room air temperature to rise, increased the ceiling panel-heat pickup, and reduced the flow of heat from the floor and walls into the room. Data in Table 1 indicate that the magnitude of these effects decreased with reduction in temperature difference between the infiltration air and the AUST. No significant changes were found with infiltration air at 80 F. In most tests, the temperature of the air leaving the room was within 1 deg of the air temperature at the 5-ft level at the center of the room.

The changes in panel-heat pickup and room air temperature caused by infiltration are of the same magnitude as those found in a floor-panel-heated room for the same infiltration rate and temperature difference between the infiltration air and AUST (see Fig. 3 of reference 2). Good agreement between the observed values and those calculated by inversion is shown in Table 1 for tests with infiltration.

*Effect of Low-Temperature Heat Sources:* The effect of low-temperature convective heat sources, such as motors and people, on the performance of a cooled ceiling panel was studied by placing 500-watt electric heaters in the otherwise empty room. Each heater consisted of a 500-watt incandescent lamp placed inside vertical, concentric 8-in. and 10-in. galvanized ducts 2 ft long and baffled at both ends. The heaters were mounted about 1 ft above the floor. The temperature of the outer cylinders was less than 100 F and calculations indicated that at least 95 percent of the total output of the heaters was by natural convection.

Tests were made with a uniform environment of 85 F AUST, 70 F ceiling temperature, no infiltration, and with two, four, or six of the heaters in operation. The effects of the internal heat sources on heat flow and air temperature are shown

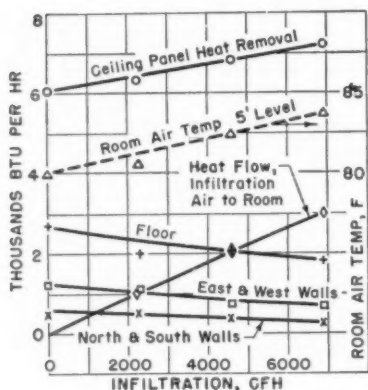


FIG. 4. EFFECT OF INFILTRATION ON HEAT FLOW AND ROOM AIR TEMPERATURE; CEILING 70.5 F, AUST 85 F, INFILTRATION AIR 110 F, UNIFORM ENVIRONMENT,  $24\frac{1}{2} \times 12 \times 8$  FT ROOM (ONE AIR CHANGE PER HOUR = 2400 CFH)

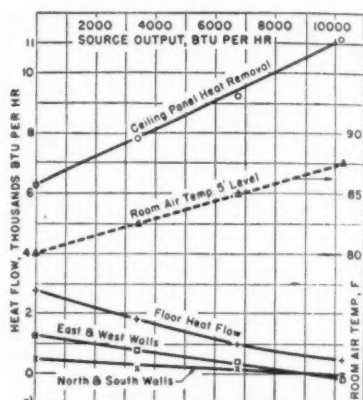


FIG. 5. EFFECT OF CONVECTIVE HEAT SOURCE ON HEAT FLOW AND ROOM AIR TEMPERATURE; CEILING 70 F, AUST 85 F, UNIFORM ENVIRONMENT, NO INFILTRATION

in Fig. 5. The top curve of the figure indicates that the heat pickup by the cool-ceiling panel increased by 50 percent of the heater output. The air temperature at the 5-ft level increased, and at the higher internal loads was higher than the controlled AUST. Because of the increased air temperature, heat flow from the floor and wall surfaces decreased, and above approximately 9000 Btu per hr internal load, the heat exchange at the wall surfaces reversed direction.

The heaters in the room produced localized, turbulent, convection currents and large temperature gradients. Directly over the heaters, air temperatures of approximately 110 F were measured while the general room temperature was about 85 F. Because of this turbulent condition, the curve in Fig. 5 of room-air temperature at the 5-ft level may not represent the true average temperatures.

Because of the similarity between Fig. 4, showing the effect of infiltration, and Fig. 5, the effect of internal heat loads, further comparisons were made. It was found that the additional heat picked up by the ceiling panel was essentially the same, whether it was caused by an internal convective heat source or by infiltration air introduced at the same temperature as the air leaving the convective source.

#### FACTORS THAT CAN BE NEGLECTED IN DESIGN

*Effect of Non-Uniform Environment:* It was shown in an earlier paper<sup>3</sup> that the performance of a panel-heating system in a space having a non-uniform environment could be predicted with satisfactory accuracy on the basis of the AUST of the space. The close agreement between observed and calculated heat transfer

rates and air temperatures shown in Table 1 suggests that this relationship is also true for the panel-cooling systems.

*Effect of Room Furnishings:* As indicated in Column 1 of Table 1, several tests were made with room furnishings, consisting of a davenport, five office chairs with leather backs and seats, one large table, and two end tables. This amount of furniture was judged to be representative for a living room of comparable size. The temperatures of the furniture, as measured on the table tops, and seats and backs of chairs, were about the same as the room air temperature and thus lower than the AUST. The ceiling panel was therefore seeing a lower temperature environment than with empty room conditions, and consequently, the heat pick-up by the panel was less. The reduction averaged about 5 percent, which is about the same as the reduction in the output of a heated floor panel caused by a like amount of furniture in earlier tests.<sup>3</sup>

*Effects of Non-Uniform Panel Temperature:* One test was made with only a part of the ceiling area cooled. The temperature of the uncooled sections was not controlled, but was the equilibrium temperature resulting from heat exchange with other room surfaces and the room air, heat gain or loss from the rear of the panel, and conduction from piping and adjacent panel sections. The area-weighted average rate of heat removal from the entire ceiling was essentially the same as the calculated heat removal based on the area-weighted average ceiling surface temperature. The observed room-air temperature agreed well with the air temperature calculated by the inversion principle when using the same average temperature of the entire ceiling. An earlier paper<sup>4</sup> indicated similar performance for heated floor and ceiling panels at non-uniform panel temperatures.

#### FUTURE RESEARCH

Additional experimentation with other sizes of rooms, and other types of internal heat sources would be desirable to develop general design formulas. Conditioned air, electric lighting, and solar energy are factors which have not been included but which are important to air conditioning. Consequently, the second phase of the overall research program, now in progress, will measure the effect of conditioned air on the performance of a cooled ceiling panel. The third and fourth phases cover the modifying effects of luminaires and solar radiation which are important air conditioning loads.

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2. ASHVE Research Report No. 1490—Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 495).
3. ASHVE Research Report No. 1499—Effects of Non-Uniformity and Furnishings on Panel Heating Performance, by L. F. Schutrum and C. M. Humphreys (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 121).
4. ASHVE Research Report No. 1516—Effects of Room Size and Non-Uniformity of Panel Temperature on Panel Performance, by L. F. Schutrum and J. D. Vouris (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 455).

## DISCUSSION

H. B. NOTTAGE, Encino, Calif. (WRITTEN): Would the authors present a complete fundamental treatment of the use and limitations of AUST?

E. F. SNYDER, JR., Minneapolis, Minn. (WRITTEN): A review of the subject paper indicates that the authors have done a splendid job in presenting the information they have obtained and supporting calculated data with experimental data. They are to be congratulated on this contribution to the knowledge of air conditioning.

It is noted that this paper constitutes the first contribution in a program which includes:

1. Heat exchanges that take place in a ceiling panel-cooled space under various conditions of surface temperatures, infiltration and furnishings.
2. The effects of the introduction of conditioned air.
3. The effects of luminaires.
4. The effects of solar radiation.

The above program appears to be comprehensive but seems to the writer to leave some areas that are in doubt.

In any application work, the parameters must be established so that the engineer does not inadvertently overstep the boundaries of good design practice. The paper in part proves the soundness of the theory of inversion and it might be assumed that this theory holds true for all conditions. But has this proven to be true? What design limits must be imposed on water temperature, tube spacing, surface temperature and panel construction?

One factor which seems to be important in panel cooling and not in panel heating is the dew-point of the air in the space and surrounding the panel. Several instances have been noted in the past where the panel has been operated at surface temperatures below the dew-point of the surrounding air with resulting formation of condensate on the panel surface.

The paper shows only dry bulb temperatures but I should like to point out that under the conditions outlined in Fig. 3, a relative humidity of 50 percent would mean that the panel is operating at a surface temperature, with 80 F air, of only 9 deg above the dew-point of the air. If this same relative humidity prevailed for 85 F air, then we find that the panel surface temperature is only 4.6 deg above the dew-point. If the relative humidity should rise to 60 percent with an 85 F dry-bulb, then the surface temperature is below the dew-point temperature.

Again, let me extend my congratulations to the authors.

J. R. CARROLL, Urbana, Ill., (WRITTEN): It is with great pleasure that we greet one of the first basic research papers presented on panel cooling, and I note with some satisfaction that the words *radiant cooling* have been omitted from reference. The term *radiant* was so mutilated in common reference to panel heating that by the *inversion principle* we should expect an equal abuse in panel cooling. However, at the risk of appearing to contradict myself, may I say that I feel radiant energy exchange may have a much more important part in panel cooling design than it ever did in panel heating.

The subject of panel cooling is much more fascinating and complicated than panel heating because of the possible condensation on the panel surface, and because of the lighting load and roof load (and other heat gain sources) that must be considered. With panel cooling, the radiant energy transfer from electric lights, hot roofs, and the direct, diffuse, and reflected solar radiation all form an important part of the ability of a panel cooling surface to perform. Also, the effect of non-steady state heat transfer to and from the panel may very well contribute to its primary importance, and will certainly complicate the energy exchanges.

The technical paper presented by Mr. Schutrum and his colleagues deals with steady state heat transfer only and panel surfaces above the room air dew-point temperature.

All of my field design and investigative work has been done with surfaces below the room air dew-point temperature, and dehumidified air is introduced into the room after passing through the panel. These warm air and cool air panel heating and panel cooling systems have an inherent self-balancing characteristic in that any heat not transferred by the panel surface will be transferred by the air introduced into the room. Tests indicate that the air entering the room will be about 70-72 F when the room air is 75 F, which indicates that a floor panel surface will absorb approximately 70-80 percent of the cooling load and the introduced air the other 20-30 percent. The total does represent 100 percent of the cooling load, no matter what its value, since the panel surface temperature is not restricted by condensation difficulties.

The reference to Fig. 3 in the text says that the panel will remove 21.6 Btu per (hr) (sq ft) from a room at 80 F with the panel at 69 F. This value of 21.6 is reasonably close to what might be expected in residential installations. If the room load were less than 21.6 and if the panel surface were maintained at 69 F, the room air temperature could be less than 80 F. However, if the room load were higher than 21.6 Btu per (hr) (sq ft) and the panel were maintained at 69 F, the room air temperature would increase above 80 F. Fortunately, the roof heat gain is absorbed by the ceiling panel directly and hence serves to reduce the room cooling load on the panel surface. I do not know why the authors chose 69 F specifically as the ceiling panel temperature to illustrate the use of Fig. 3, but no matter what value is used it is obviously a critical design value that needs an extensive amount of further study. The determination of the limiting panel temperature before condensation, and methods of control over this temperature, are absolute necessities for the design and use of closed panel cooling systems. Due to the large number of existing ceiling panel heating systems, it appears that a great deal of interest could be created in ceiling panel cooling systems with the use of separate dehumidifiers to reduce and to control humidity. It should be noted that there are also many floor panel heating systems in existence and the application of cooling to them is also of importance to the industry and profession.

I wish to congratulate the authors for their well written paper. The results of their studies have been presented in a simple and concise fashion and I am anxiously awaiting the results on the heat transfer studies in a room cooled by a panel in combination with conditioned air.

C. S. LEOPOLD, Philadelphia, Pa., (WRITTEN): In June of 1947 I presented to the ASRE the first of a series of papers on panel cooling\*. The summary stated in part: "Engineering for panel cooling is not the simple reverse of panel heating. Neither the concept of combined surface conductance, nor of the grey body are adequate for the cooling analysis."

The conclusions of the Laboratory paper state: "Ceiling-panel cooling is an inversion of floor-panel heating, and the performance of a cool ceiling panel can be predicted from the performance of a warm floor panel."

By custom, and for reason, we design heating for an empty building, at night, without illumination. We design cooling for an occupied building, in the daytime, with artificial light and sunlight. In my judgment therein lies the error in the Laboratory statement.

For example, with filament lighting the temperature of the radiating source is approximately 4800 F. and a large portion of the radiant energy is transferred directly to a structure by radiation. The energy represented by this portion of the radiation can be carried away by a ceiling panel with water circulation without depressing the panel surface below room air temperature. Furthermore, the portion of the high temperature radiation which falls on the other room surfaces tends to raise their surface temperature and this facilitates transfer to the cooled ceiling by radiation. The percent

\* The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by C. S. Leopold (*Refrigerating Engineering*, July 1947, p. 33) and The Mechanism of Heat Transfer, Panel Cooling and Heat Storage—Part II—Solar Radiation, by C. S. Leopold (*Refrigerating Engineering*, p. 571).

of the source energy which is removed without depressing the panel temperature I called *Independent Radiation Transfer*, or *IRT*. This factor varied from 1.5 to 37 percent, depending on the type of luminaire and the type of ceiling paint used in our tests.

Similar statements on IRT, too long to repeat here, can be made for the solar load.

In a complete panel cooling system, air is supplied for ventilation and humidity control. If the dehumidification is accomplished by refrigeration, cooling of the air is required; if the cool air is introduced into the room it will provide a portion of the cooling effect.

Assume for a moment that the IRT is 20 percent and that 40 percent of the cooling is accomplished by the ventilation air. The area of the panel and its temperature depression need only be that required to do 40 percent of the cooling load. By neglecting the IRT the panel area and its temperature depression as calculated would be for 60 percent of the cooling load or 50 percent greater than required.

This is not a criticism of the excellent experimental work covered by this paper but I do find fault with the conclusion. As a minimum the conclusion should be corrected to read: "Ceiling panel cooling for the transmission load . . ." etc., or, to be somewhat facetious, "Ceiling panel cooling for an empty building at night . . ."

A. J. HESS, Los Angeles, Calif., (WRITTEN): The most interesting observation to be made concerning the paper on *Radiant Cooling* is that the society has at long last been able to start on research work having to do with radiant cooling as an adjunct to air conditioning.

Previous work and installation results reported to us by Leopold, Wiggs, Day, and others have established the fact that usual space occupied by air conditioning systems can be greatly reduced by the use of radiant cooling, an increasingly important consideration of design as costs of constructing building space continue to skyrocket.

It is indeed too bad that the society lacked the funds and facilities to catch Mr. Leopold's forward pass of a few years ago on radiant cooling, for we would probably be approaching the goal by now instead of just starting our work.

This paper is preliminary in character, but does state the welcome fact that much of the criteria previously established for panel heating can be used in the cooling research to save much time, effort and research funds. Thus the work can get on more rapidly to the more important study of the effect of radiant panel cooling on luminaire and solar heat gains. I congratulate the authors on the work and hope it continues as rapidly as possible.

P. B. GORDON, New York: Again, the authors are to be commended for their excellent work. The TAC on Panel Heating and Cooling are proud of this series of papers by the Laboratory, with this the first on the problem of room heat exchange with cooled surfaces.

One interesting phase of the paper that requires further study has to do with the split-up of radiation and convection exchange. In the inversion picture given in Fig. 1, the values are based on a temperature difference of 20 deg for AUST to ceiling temperature and a value of 13.6 deg for room air temperature to ceiling temperature. If the heat pick-up is divided into radiation exchange and convection exchange, we would come up with something like a radiation exchange coefficient of 1.0 Btu per hour per square foot per degree difference, and a convection exchange of about 0.6 Btu per hour per square foot per degree difference. For this particular example, the overall exchange based on air to ceiling temperatures worked out to about 2.06 Btu per hour per square foot per degree difference between the ceiling temperature and the room air.

In the same way for the information shown in Fig. 5, wherein values are given on AUST for three particular conditions,—with varying amounts of excess convection heat,—while the radiation coefficient should be the same the convection coefficient increased from about 0.5 to about 1.24.

Yet—and this is interesting though I am at a loss to explain it—if you use air tem-

perature to ceiling temperature differences for all three conditions, the overall heat exchange coefficient ends up at 2.0 or 2.1.

In Fig. 3, where the AUST values are not given but certainly have some influence in the output, if you use ceiling to air temperature difference only, the overall heat exchange again ends up with a unit value of 2.0 to 2.1.

I believe this particular region requires further study to determine the significance of this particular relationship.

This is an aside on something that Mr. Leopold mentioned. His work and the information that he has developed in the many years that he has been working with panel heating and panel cooling have been interesting and always of great value.

However it must be remembered that these particular experiments reported on by the authors are heat exchange experiments in a laboratory under controlled conditions, and are not simulated conditions encountered in practice. Therefore, the data and the interpretations should be recognized as such.

These data did not result from field or from mathematical analysis. These studies involved controlled measurements in a controlled laboratory and operated deliberately for this purpose. I believe this may explain why the authors were interested in emphasizing the *inversion* theory for the particular conditions of the paper, which definitely would not hold under the influence of luminaires, solar energy, thermal storage and conditioned air that come into play in a normal room in practice.

A. T. JORN, North Chicago, Illinois: This paper is certainly welcomed by everyone who has been in this field for very long. It finally provides unbiased information in a clear-cut manner.

It is interesting to note on Fig. 3 that the example the authors have presented comes out 21.6 Btu per square foot; in private research done elsewhere (but certainly not done as carefully) it came out 21 Btu under identical conditions. The fact that the data are that close is certainly reassuring to all investigators involved.

One thing that bothers me a little about this paper, however, is the format of Fig. 3. If you will look at it a moment you will see that the authors have presented the heat removal from a panel for a specified condition of room air temperature and ceiling panel temperature. I feel that presentation can be misinterpreted very readily.

You realize that what they have said in effect is that in order to have, under steady state conditions, a ceiling panel of a given temperature (69 F) and simultaneously room air of a given temperature (80 F) the AUST is already fixed; that it can only be one definite value because the only heat that is coming into the room is coming from the wall surfaces and they must be at a certain average temperature in order to supply heat at exactly the same rate that the panel is removing it. The implication is, I believe, a little too subtle for printed research data.

What I am trying to point out is that if a particular room were two to three times as high there would be approximately twice the surface contributing heat to the room; and therefore there could be a much lower AUST and still contribute the same amount of heat. With the AUST lower, there would not be as much radiant transfer between the walls and the ceiling panel so a panel of a given temperature in a room of a given air temperature would not absorb as much heat as in the case of the test room.

Even in a room with a very high ceiling, this fact might not be too consequential in the case of ceiling panel cooling due to the relatively small percentage of radiant heat exchange. In the case of a ceiling panel heating, however, with 90-95 percent of heat exchange by radiation, the effects could be substantial.

In the case of a practical application where part of the cooling is usually done by cool air, the AUST could be a great deal warmer than was the case in the test room so that the panel pick-up would be much higher than Fig. 3 would indicate for design conditions of 80 F room air and 69 F average panel temperature. The AUST, therefore, would have to be calculated to allow any reasonable approximation of actual panel pick-up; so the data should include AUST as a variable.

Thus it can be seen that whereas the data of Fig. 3 are true in every sense of the word, use and interpretation must be handled very carefully.

One other point I would like to bring out is in connection with the paragraph on effects of non-uniform panel temperature. In private research that has been unpublished to date, essentially the same thing has been found, i.e. that a non-uniform cooled or heated ceiling panel will have essentially the same effect as if it were at an uniform surface temperature equal to the area weighted average panel temperature.

However, it was also found that (for some reason as yet unexplained) if the non-uniformly heated ceiling panel is above a certain average temperature (higher than the range of the tests reported here today) one gets panel outputs that are substantially higher than with an uniformly heated panel at the same average temperature.

P. R. ACHENBACH, Washington, D. C.: I want to comment and check with Mr. Schutrum about one statement that he made in his presentation with regard to the effect of infiltration on heat transfer.

If I read the last three tests under steady state conditions right, in Table 1, they used three different rates of infiltration, one, two and three air changes substantially, and in this case the air was brought in at 80 F, or equal to the room air temperature; the AUST was substantially constant, the air temperature was substantially constant for those three tests, and the heat transfer was substantially constant.

Therefore it seems to me the conclusion to be drawn would be that the air motion produced in the room, due to the difference in air changes, had little effect on the heat transfer.

Perhaps the way in which the infiltration air was brought in might be significant and perhaps the conclusion that would be drawn from this is that the additional air motion due to greater infiltration as being not effective might not always apply, depending on where the infiltration air came in.

The point is the amount of air didn't seem to make any difference in the heat transfer when the infiltration air was at room temperature.

I believe this paper will be helpful to those who may be considering the possibility of using panel cooling for residential application. Most engineers have steered away from that, I think, perhaps because of the condensation problem.

It seems to me that in order to design a residential system, it would be almost necessary to combine it with forced ventilation of the house; and I can envisage a sort of two-stage system in which a dehumidifying coil is used to bring in the ventilation air and slightly pressurize the house so there would not be random infiltration causing local hazard of condensation and using that as a first stage of cooling when rather low temperature differences prevail between outside and inside, but might have high latent load, and then as the outdoor temperature became higher one would supplement the cooling and dehumidification by panel cooling from a ceiling.

It seems to me this paper will give engineers a start at least on the design of systems for panel cooling of residences.

I think it also points out that a ceiling panel of high thermal conductivity, such as metal plates with the tubes clipped to them, would have an advantage over a panel that had coils imbedded in plaster because you would have less temperature difference between surface and water temperature.

LESTER T. AVERY, Cleveland, Ohio: When a paper is presented that is emphasized, as Mr. Gordon emphasized, that it is a theoretical laboratory test only and yet has tremendous value from the standpoint of practical application, I think the relationship could well be pointed out.

I think it was Dr. Giesecke some years ago who emphasized the point that the ceiling doesn't care how it got that way. The ceiling doesn't care whether it became cooled or heated from a direct source of energy, or whether it got it indirectly from someone blowing air across it.

This method of radiant cooling and panel cooling was over-emphasized some years ago by, I believe, Dr. Mills who thought that you could do everything with the radiant panel in absorbing heat and because of the over-emphasis it was discredited.

We know now that the ceiling panel, no matter what material, whether it be a perforated metal or a luminous plastic, or a simple ceiling arrangement with horizontal air flow patterns across it, can use this principle of first reducing the ceiling temperature to offset the other higher temperature which comes from the luminaires; and second, to actually go part way in effectively cooling the room.

The next step I was hoping Charlie Leopold would emphasize is that these tests be continued to show the relationship in what would be called the effective temperature, when we have changed the pattern in the room from a warm ceiling to a cool ceiling.

We know that radiant effect is very important in the feeling of comfort. An exaggerated case of that is the hot sun load which is absorbed by a venetian blind, which immediately changes the requirements in the room as compared with the basement room where there is no sun load. The same thing is true of the high lighting intensities; and I am told now that good lighting is going to go to 8 or 9 watts per square foot even with fluorescent lighting, and that means we have a heat source which, whether we want it or not, becomes a panel heater.

This paper is certainly of the utmost importance for those who are thinking in terms of simply having a radiant problem by using panel cooling or radiant cooling. I think panel cooling is a better phrase. I think we can understand panel cooling better than radiant cooling.

But anyhow, the point that I emphasize is again, as Dr. Giesecke pointed out, the ceiling does not care how it got that way.

**AUTHORS' CLOSURE (Mr. Schutrum):** Mr. Snyder has discussed the importance of preventing condensation in the operation of cooling panels. The purpose of the tests reported here was to determine the heat transfer to the cooled panel, and in all tests the dew point was controlled so there was no condensation. It is regrettable that a statement of this fact was not included in the paper.

Mr. Leopold's objection to the statement that ceiling panel cooling is an inversion of floor panel heating is valid for the conditions he has stated. It is not an inversion if solar energy lights, occupants and other internal heat sources are included, but only within the limitations stated in the report. Perhaps it was not emphasized enough in the paper that the inversion principle applied only to this preliminary or phase-one study under controlled laboratory conditions, which excluded the effect of lights and solar radiation.

Mr. Gordon has pointed out that the overall coefficient of heat transfer from room air to the panel is of the order of 2 Btu per (hour) (square foot) (Fahrenheit degree) temperature difference. Actually, Fig. 3 was computed using a convection coefficient of  $h_c = 0.31 \theta^{0.31}$ . This is the convection coefficient which was established for the room heated by a floor panel and under the inversion principle applies to the cooled room. When the temperature difference between the panel and AUST is about 10 to 20 deg F., the combined radiation and convection transfer based on the temperature difference between the panel and the room air becomes about 2. For lower temperature differentials the combined coefficients are smaller than 2 and for higher differentials they are larger. With infiltration or internal heat sources in the room the overall coefficient remains about the same, but a convection coefficient other than  $h_c = 0.31 \theta^{0.31}$  exists.

A complete report will be made in another paper, giving the convection and radiation coefficients measured in the Environment Laboratory.

In answer to Professor Carroll's interesting comments, the choice of 69 F as a ceiling temperature to illustrate the use of Fig. 3 was arbitrary and has no particular significance.

In Mr. Achenbach's discussion, mention was made that various infiltration air change rates at 80 F had no effect on the room air temperature nor the panel pickup. This is shown in Table 1 for tests Nos. 387, 388, and 389. Re-examination of the original data for these three tests showed that the heat transferred to or from the infiltration

air was small (maximum 41 Btu per hr). The only other effect that the infiltration could have would be to change the convection from the room surfaces because of increased air motion. The data showed, however, that this increased air motion did not cause a significant trend in heat flow for any of the six surfaces. Consequently, the increased turbulence caused by infiltration rates up to three air changes per hour must not be great enough to cause a significant change in the convection transfer. This does not hold, moreover, if the infiltration is at a different temperature from the room air as shown by the three tests in Table 1 (391, 392, and 393) with the infiltration air at a higher temperature than the room air temperature.

Dr. Nottage asked for a complete fundamental treatment of the use and limitations of AUST. The AUST is defined as the area-weighted average temperature of the uncooled room surfaces, which for this report are the four walls and the floor excluding the ceiling panel which was cooled by liquid circulation.

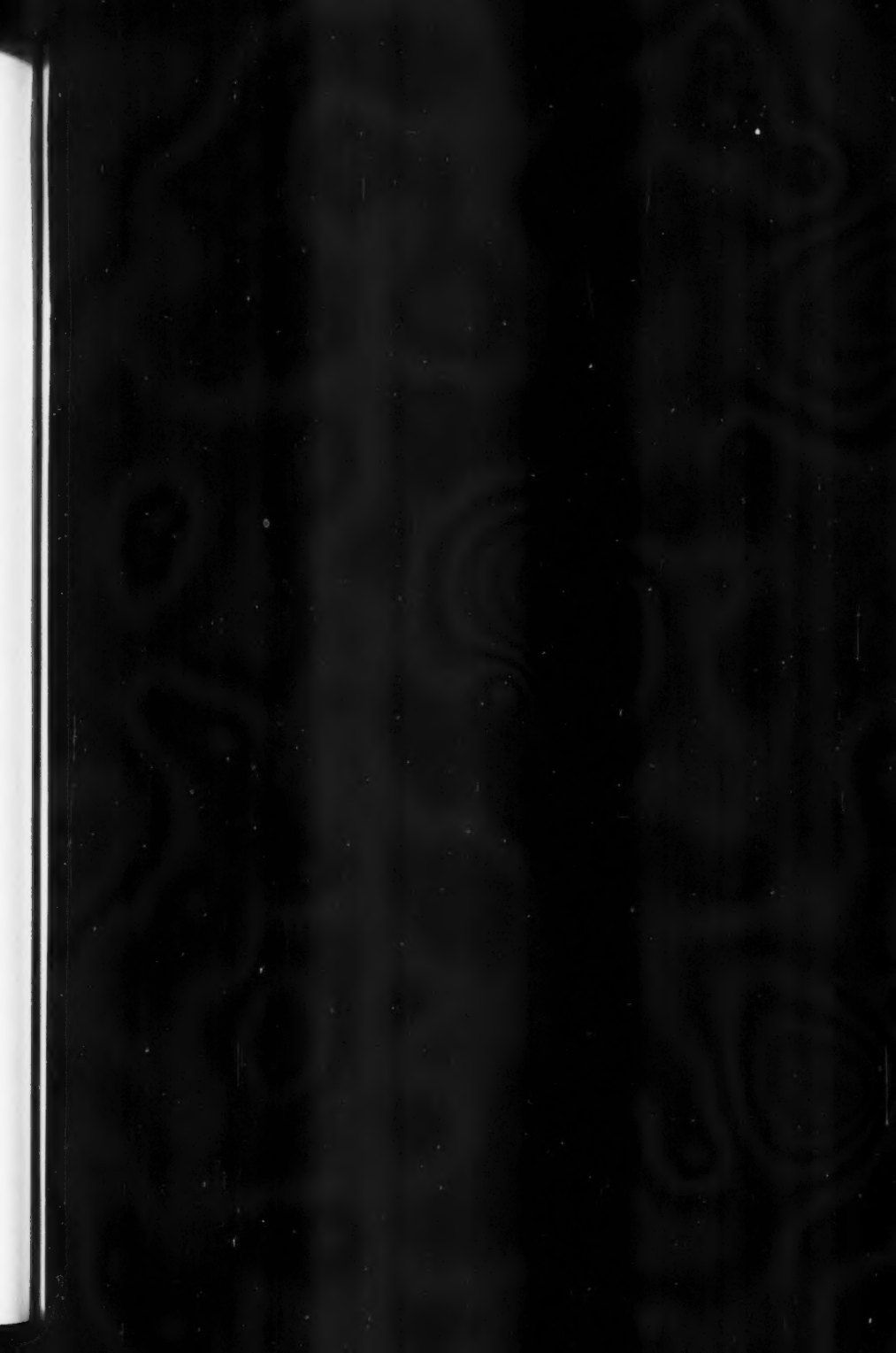
The AUST is therefore a means for representing a complex surface temperature pattern by one equivalent surface temperature which in turn serves to simplify the determination of radiation and convection exchanges in the room, and particularly the panel heat exchanges. The AUST must be both an equivalent convection temperature and an equivalent radiation temperature but because the laws which govern radiation and convection transfer are different, at most the AUST is a compromise. Within a given mode of heat transfer there may be differences. For example the radiation exchange per unit area between the ceiling and the floor may be different from the exchange between ceiling and a wall because of shape factor or seeing factor differences, yet the AUST weighs the two surfaces equally. Radiation exchanges are proportional to the difference in absolute temperatures to the fourth power and to the surface emission characteristics but no adjustments are made for this in the AUST. Convection, on the other hand, behaves differently for floor positions than it does for the wall position and is not a linear function of temperature but these differences are not accordingly weighed in the AUST.

The AUST is, in effect, an engineering tool which for the conditions usually found in the home will give engineering accuracies of the order of 5-10 percent under steady state conditions. For unsteady state conditions, where thermal storage becomes important, the use of the AUST would be limited to instantaneous heat transfer determinations.

Mr. Jorn stated that in private research a heat transfer rate of 21 Btu per (hr) (sq ft) versus 21.6 reported here was found for the same conditions as shown in the illustrated example. It is always gratifying and reassuring to know that different researchers obtain the same result. It is equally important to know when the data are not compatible.

Mr. Jorn's discussion of Fig. 3 emphasizes the fact that the AUST which is not given in the figure, nevertheless, is related directly to the panel pickup, panel temperature, and room air temperature. For a given room, there can only be one AUST when the other three variables are fixed. Other factors such as infiltration, and internal loads would change this relationship.

Mr. Jorn states that if the area of the AUST were doubled, a much lower AUST would contribute the same amount of heat to the room by convection; and with the lower AUST the radiation heat exchange between the cooled panel and the AUST would be lower. This is technically correct and one would suspect that Fig. 3 would not apply for room differing in size from the test room and especially so when the heat transfer is essentially radiative as for a room heated by a ceiling panel. Nevertheless, calculations show that the variations in heat exchange are not appreciable and indicated that Fig. 3 would apply within engineering accuracy for the uniform environment condition and ceiling heights up to 12 ft. Data obtained in a room heated by a ceiling panel show that for the relationship of room air temperature, panel temperature, and panel output similar to Fig. 3, (uniform environment) the panel output with the ceiling height of 12 ft, differed by less than 5 percent from the output when the ceiling height was 8 ft. The data for these tests can be found in reference 4.



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## MEASUREMENT OF ANGULAR EMISSIVITY†

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This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

IN PROBLEMS of heat transfer dealing with the exchange of heat radiation, it is a common practice to use the normal (perpendicular) emissivity values of the surfaces involved. It is assumed that the emissivity of a surface element is the same regardless of the direction of emission into the surrounding hemisphere and that it is numerically equal to the normal emissivity value. In most cases, this assumption is justifiable and gives sufficiently accurate results. However, it has been established from theory and verified by experiment that the emissivity of a surface element varies as the direction of emission departs from normal. Some problems of heat transfer, particularly those encountered in research work, require a knowledge of this variation of emissivity with angle. From a practical viewpoint it may be necessary to give consideration to the directional characteristics of radiant panels, such as small high temperature panels.

This paper describes apparatus for angular emissivity measurements and reports values obtained for a number of surfaces. Most of the surfaces were painted ones, because no previous data for such surfaces were available.

The investigation was initiated as part of the research carried on at the ASHAE Research Laboratory under Group B of the TAC on Panel Heating and Cooling.□ Data were needed for the painted surface of the Environment Laboratory and for globe thermometer paints. Emissivity values were also needed for two other projects being carried on at the Laboratory.

### APPARATUS

Essential features of the apparatus were: (1) a sensitive radiometer with its field of view restricted to a narrow cone angle (see Figs. 1 and 2), (2) calibration apparatus for the radiometer, (3) a means of heating the sample and holding it at the desired angles with respect to the line of sight of the radiometer, and (4) a means

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□ Based upon a thesis submitted by Aydin Umur to Case Institute of Technology in partial fulfillment of the requirements for his M.S. degree in Mechanical Engineering.

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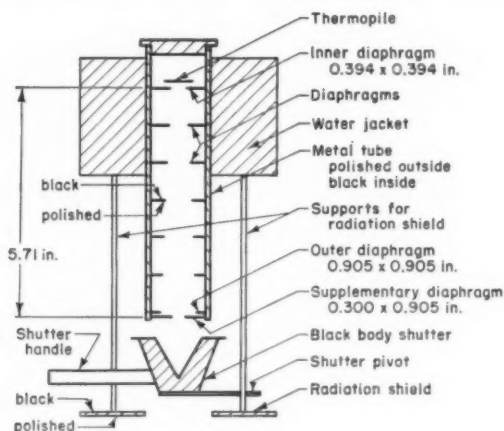


FIG. 1. SCHEMATIC DRAWING OF RADIOMETER

of controlling and evaluating the environment surrounding the sample so that full account could be taken of stray radiation. The radiometer was a commercial instrument to which modifications were made as described in Appendix A.

*Test Surface Preparation and Shielding:* All test surfaces were prepared on flat metal plates 6 x 12 in. in size and from  $\frac{1}{16}$  to  $\frac{1}{8}$  in. in thickness. To measure the temperature of the test surface eight thermocouples were installed in grooves cut into the back surface of the plate. A heater unit was bonded to the back surface of this plate with a double-sided adhesive tape. It was covered with a  $\frac{1}{2}$ -in. thick insulation board to avoid excessive heat loss to the surroundings and to insure uniform heating, which is an important factor, since the radiometer sighted a large area at high angles. The heater element was made from 0.002-in. thick constantan foil. It was designed for 24 volts and had a resistance of 5.73 ohms. A film type adhesive was used to bond a layer of anodized aluminum foil on each side of the element. Details of a similar bonding process have been previously described.<sup>1</sup> The heat input was sufficient to obtain sample temperatures up to 180 F and was regulated by means of an auto-transformer and a voltage stabilizer.

To shield the sample from stray radiation, it was placed inside a cylindrical tank which was in turn placed inside a slightly larger rectangular tank (see Fig. 3). A pump circulated water in the space between the two tanks to maintain the cylinder at a uniform temperature. The sample was supported in the cylinder at the lower end of a vertical rod. The upper end passed through a bearing fixed in a rigid frame and carried a protractor. A sight tube in one side of the tank was provided so that the radiometer could see the sample. The inside surface of the cylinder was blackened with paint which had a hemispherical emissivity of 0.90. This arrangement provided an environment of known temperature and emissivity so that reflected radiation could be accounted for as described in Appendix B.

<sup>1</sup> Exponent numerals refer to References.

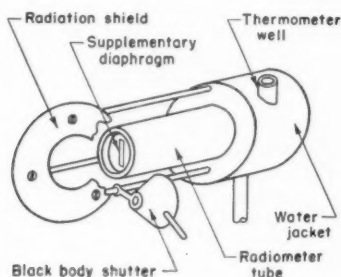


FIG. 2. EXTERNAL VIEW OF RADIOMETER

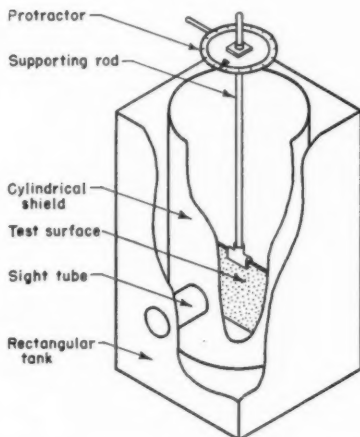


FIG. 3. METHOD OF SHIELDING TEST SURFACE FROM STRAY RADIATION

All temperatures were measured with 30-gage copper-constantan thermocouples made from specially calibrated wire.

#### TEST PROCEDURE

The radiometer was first calibrated by the procedure given in Appendix A and was subsequently checked at intervals throughout the period of the tests. Determination of emissivity required a measurement of the radiometer output while the black body shutter covered the radiometer tube, a measurement with the shutter removed so the thermopile could see the sample, and finally a second measurement with the shutter covering the tube. Temperatures of the sample, the cylindrical shield surrounding the sample, the radiometer shutter, and the radiometer were also recorded. These readings were repeated for each angle at

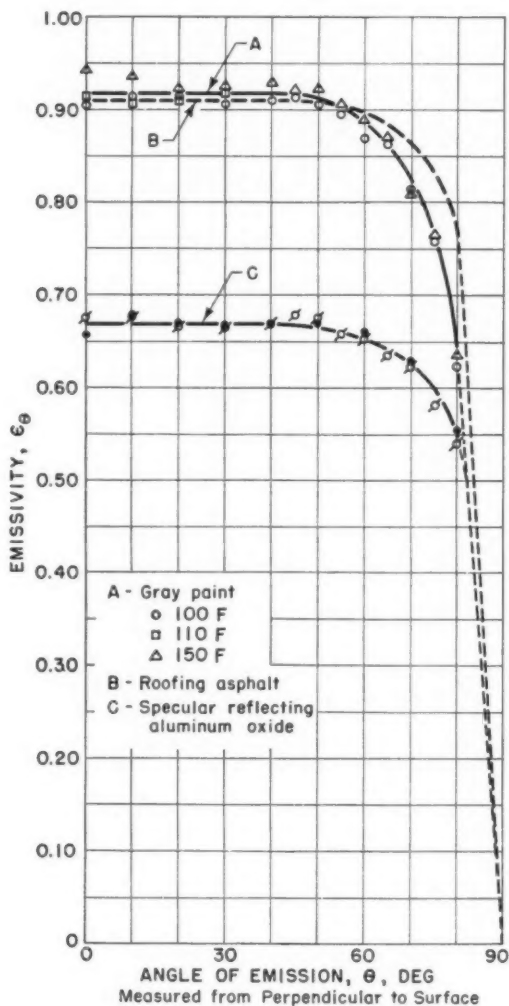


FIG. 4. EMISSIVITY vs. ANGLE OF EMISSION FOR GRAY PAINT, SPECULAR REFLECTING ALUMINUM OXIDE AND ROOFING ASPHALT

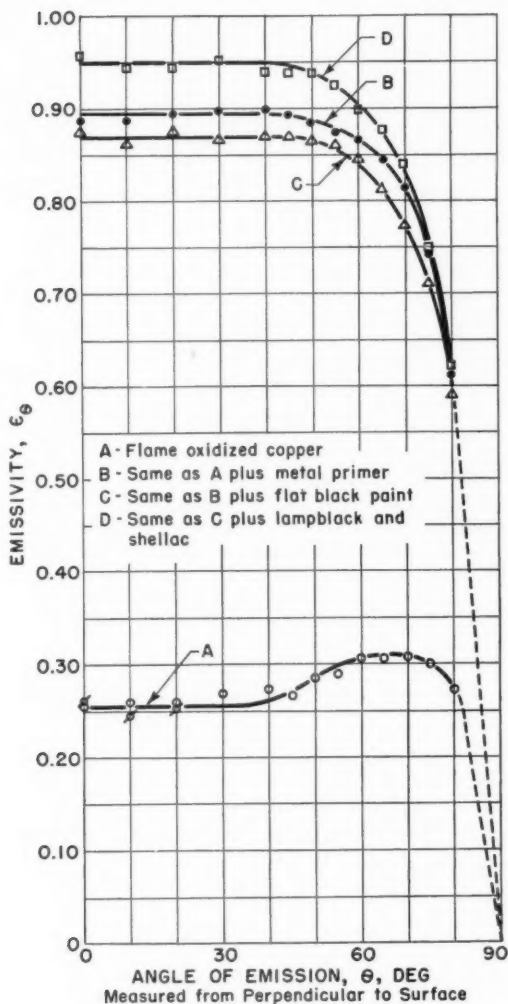


FIG. 5. EMISSIVITY vs ANGLE OF EMISSION FOR OXIDIZED COPPER COATED WITH SEVERAL DIFFERENT PAINTS

which the sample was viewed. Reduction of the data, including corrections for reflections and temperature drop across the sample thickness, is described in Appendix B.

### RESULTS

The variation of emissivity with angle was determined for the surfaces listed in Table 1, which gives the total hemispherical emissivity and the normal (perpen-

TABLE 1—NORMAL AND HEMISPHERICAL EMISSIVITY OF SURFACES

No.	SURFACE DESCRIPTION	$\epsilon_n$	$\epsilon_h$
1	1/8-in. thick aluminum plate with two coats of reddish brown primer <sup>a</sup> and one coat of light gray paint <sup>b</sup> , all coats sprayed on . . . . .	0.92	0.88
2	1/8-in. thick copper plate, flame oxidized . . . . .	0.26	0.27
3	Same as No. 2, with one coat of reddish brown primer <sup>a</sup> added, brush applied . . . . .	0.90	0.85
4	Same as No. 3, with one coat of flat black <sup>c</sup> added, brush applied . . . . .	0.87	0.83
5	Same as No. 4, with one coat of lampblack <sup>d</sup> added, brush applied . . . . .	0.95	0.90
6	Weathered roofing asphalt (melting at 180 F approx.) on 1/8-in. thick aluminum plate . . . . .	0.91	0.85
7	1/8-in. thick aluminum with anodized coating $11 \times 10^{-5}$ in. thick; specular reflecting surface . . . . .	0.67	0.65

<sup>a</sup> Reddish brown primer			
Pigment (60% by weight)	% by weight	Vehicle (40% by weight)	% by weight
Zinc chromate	20	Vegetable oils	24
Red lead	25	Varnish	50
Zinc oxide	15	Japan drier	15
Iron oxide	20	Mineral spirits	11
Magnesium sulphate	20		
<sup>b</sup> Light gray paint	% by weight		
Titanium dioxide	6.5		
Titanium calcium pigment	39.5		
Soya-alkyd resin	24.0		
Mineral spirits, driers	30.0		

<sup>c</sup> Flat black enamel paint

<sup>d</sup> Lampblack mixed with 4-lb cut shellac and thinned with alcohol.

dicular) emissivity. Figs. 4 and 5 show the variation in emissivity,  $\epsilon$ , with angle of emission,  $\theta$ . The hemispherical emissivity was computed in the following manner. Any radiating surface element emits radiation through a hemispherical envelope. If this envelope is divided into spherical zones, the hemispherical emissivity,  $\epsilon_h$ , can be found by the following equation:

$$\epsilon_h = \frac{\sum_{\theta=0}^{\theta=90} \epsilon_{\theta} A_{\theta} \cos \theta}{\sum_{\theta=0}^{\theta=90} A_{\theta} \cos \theta}$$

where

$\theta$  = angle of emission, degrees, measured from the perpendicular to the surface.

$\epsilon_{\theta}$  = emissivity at angle  $\theta$ .

$A_{\theta}$  = area of hemispherical zone of unit radius corresponding to the average angle  $\theta$ .

In the computations, nine zones of 10 deg width were used.

The table shows that the hemispherical emissivity,  $\epsilon_h$ , is approximately equal to 0.95 times the normal emissivity. The values in Table 1 are based upon the curves shown in Figs. 4 and 5. All values shown in these figures were obtained with the 0.3-in. auxiliary diaphragm on the radiometer. Sample temperatures ranged from 100 to 125 F except for one run on sample No. 1 at 150 F. In this range, temperature appeared to have no significant influence on emissivity. Where necessary, all values have been corrected for temperature drop across the paint film. In several cases, measurements at low emission angles were made with 0.2 and 0.5-in. auxiliary diaphragms. The results were the same as those obtained with the 0.3-in. diaphragm.

The results are influenced by errors in measuring the radiometer outputs and the temperatures and errors in corrections for reflections. A possible error of  $\pm 0.5$  percent in the potentiometer-amplifier circuit introduces an error of about the same amount in the emissivity value. Associated with this are uncertain errors in applying corrections for the effect of water vapor absorption on the instrument constant. These corrections were determined after the measurements and the dew points prevailing in some tests had to be estimated. The maximum error in allowing for water vapor absorption probably is less than  $\pm 0.5$  percent.

The absolute value of the temperatures measured is accurate to within about 0.2 deg, the probable accuracy of the wire calibration. On a relative scale, each temperature is accurate to perhaps  $\pm 0.1$  deg so that the maximum error introduced in the emissivity value is  $\pm 1.0$  percent with the average error about half this value. The temperature drop across the paint films was estimated with an error of about 20 percent, but since the drop is small, the error introduced into the final result is small, about  $\pm 0.3$  percent.

Errors in correcting for reflections in the case of the diffuse-reflecting surfaces are believed to be negligibly small. In the case of the specular reflecting aluminum, the error is perhaps  $\pm 2.0$  percent. Therefore, considering all the errors listed, the emissivity values given are probably accurate to within  $\pm 2.0$  percent for all but the specular reflecting aluminum, for which the probable accuracy is  $\pm 4.0$  percent.

*Discussion of Results:* Schmidt and Eckert<sup>2</sup> applied electro-magnetic wave theory and the Fresnel reflection formulas to show that emissivity varies with the angle of emission. For surfaces of non-conductors, they showed that the value of emissivity stays constant between normal and 50 to 60 deg from normal, after which it begins to decrease at an increasing rate, reaching zero as a limit at 90 deg from normal. For metals, the emissivity is constant out to about 45 deg from normal, then increases until the angle reaches about 75 deg, after which it drops to zero at 90 deg from normal. Measurements for common substances by Schmidt and Eckert<sup>2</sup> are in close agreement with theory.

Figs. 4 and 5 show that the painted surfaces have the characteristic behavior of non-conductors and follow the theory developed by Schmidt and Eckert. Estimated corrections for temperature drop across the paint film have been applied to all values. The normal value for lamp black, 0.95, is in close agreement with published values. It is of interest to note that the metal primer has a slightly higher emissivity than the flat black paint.

The values for the copper oxide surface show some metallic properties, possibly because the oxide film may have been so thin that some characteristic of the copper showed through the oxide coating. Comparisons with other data for copper oxide are valueless because film thickness is an important variable and is usually not known.

The asphalt behaves as a non-conductor. Because of the thickness of the layer applied to the heating plate, special tests were made to determine the temperature drop across the layer.

The specular reflecting oxide coating (anodized) aluminum also behaves as a non-conductor, despite the thinness of the oxide coating, which was measured by others<sup>3</sup> and found to be 0.00011 in. in thickness. In regard to this, Taylor and Edwards<sup>4</sup> have studied the variation of emissivity of anodized aluminum with film thickness and found that for thickness of 0.00002 in. (3 to 5 times the thickness of a natural oxide coating) the emissivity is about 0.11. It increases rapidly with thickness until the film thickness is about 0.00008 in., for which the emissivity is about 0.63. Greater thicknesses cause very small increases in emissivity. In this instance, the measured value of 0.67 (based upon the two series of test results plotted) checks the value of 0.67 that would be predicted from the work of Taylor and Edwards and also compares well with a value of 0.64 determined by a recent measurement<sup>5</sup> in which the Taylor and Edwards apparatus was used. (An average of nine normal incidence determinations gave a value of 0.68.)

It should be noted that the light absorptance of anodized aluminum with this coating thickness of 0.00011 in. is about 0.16 (see Reference 4) as compared with its absorptance (emissivity) of 0.68 for low temperature radiation. This absorptance for light is substantially the same as that for commercial aluminum foil<sup>4</sup>, which has an emissivity of about 0.05 for low temperature radiation.

It is also of interest to point out that measurements<sup>5</sup> of the light absorptance of a number of materials and paints for angles of incidence between zero and 90 deg yield curves similar in shape to those given in this paper. For angles greater than 60 deg, the absorptance decreases rapidly and reaches zero at 90 deg. The absorptance of glass for sunlight behaves similarly.

### CONCLUSIONS

Based upon the data presented, the following conclusions can be drawn:

1. The emissivity of the painted surfaces investigated decreased as the angle of emission increased in accordance with the theory for non-conducting materials. A specular reflecting aluminum oxide coating behaved in similar fashion.
2. Hemispherical emissivity for these surfaces is very closely 95 percent of the normal emissivity.

### ACKNOWLEDGMENT

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## APPENDIX A

### THE RADIOMETER

The radiometer, a commercial instrument, was a Moll-type thermopile consisting of 66 short thermo-elements mounted inside and near the closed end of a tube (see Figs. 1 and 2). The series of rectangular diaphragms fitted in the tube minimized convection effects. The water jacket acted as a temperature stabilizer and minimized drift of the thermopile temperature.

To adapt the instrument to this work, the following techniques were employed:

1. To avoid impractically large sample sizes for emissivity measurements at high angles of emission, a diaphragm was placed immediately in front of the largest diaphragm in the radiometer tube to reduce the horizontal dimension from 0.905 in. to 0.300 in.
2. With the new diaphragm arrangement, the radiometer had an angle factor of 0.002585, *i.e.*, when sighting a black body surface, the thermopile intercepted about 0.26 percent of the total radiant energy emitted by that surface. Consequently, the output of the radiometer for the sample temperature range used was of the order of a few microvolts. The output was therefore magnified by 200 by means of a standard d-c amplifier and then measured by a precision potentiometer and a light beam galvanometer. The latter had a sensitivity of 0.47 microvolts per millimeter deflection of the light spot. Special lead wires to connect the radiometer terminals to the amplifier were essential to avoid pick-up of stray emfs.
3. It was observed that the instrument usually had a definite output when the radiometer was closed off by the shutter. This was attributed to small temperature differences between the thermopile cold junctions, the shutter, and the tubular housing which gave rise to a net radiation exchange. In order to measure accurately the contribution of the shutter and to eliminate, as described later, the contribution of the housing, the flat two-leaved shutter provided with the instrument was replaced by a small conical black body of solid aluminum. Like the shutter, it could be moved into and away from the line of sight of the radiometer.
4. To minimize radiation and convection effects from the surroundings, the radiometer housing was enclosed with aluminum foil.
5. Readings were found to be unreliable in windy weather. Adiabatic compressions and expansions of the room air due to external wind effects were found to be responsible.

The sensitivity and the response of the radiometer were found to be satisfactory. A change of about  $0.45 \times 10^{-6}$  Btu/hr in the amount of radiation falling on the thermopile area could be detected in the radiometer output and the instrument had an almost immediate response.

### RADIOMETER CALIBRATION

A blackened cone heated by a water bath met the requirements of a black body (see Fig. A-1) for calibration. The cone was of  $\frac{1}{8}$ -in. thick sheet copper with a 30 deg included angle and 4-in. diameter base. It was determined that any ray of radiation which fell on the conical surface between the apex of the cone and about 3 in. from the base was reflected at least three times. Hence, with the black surface having a hemispherical emissivity of 0.90, this section was judged to have an effective emissivity of substantially unity. The conical surface *seen* by the radiometer was restricted to this section.

The bath was heated by two flexible electrical heaters coiled around the outside of the cone. One (500 watts) was manually controlled by an auto-transformer; the other (50 watts) was controlled by a thermo-regulator in the water bath. A stirrer provided even temperature distribution. With this arrangement, the black body could be maintained constant to within 0.1 deg at any temperature up to 200 F.

Calibration was carried out as follows. The black body described previously was heated to the desired temperature. When equilibrium conditions were obtained, a series of three observations were made. The first was the radiometer output when the radiometer tube was closed by the small conical black body shutter (see Figs. 1 and 2), hereafter referred to as black body No. 1. The second was the output when the shutter was moved to one side and the instrument was allowed to see the black body heated by the water bath, hereafter referred to as black body No. 2. The third observation was the output when the radiometer was again covered by the shutter, black body No. 1. Temperatures of the black bodies and of the radiometer water jacket were also observed. The latter was substantially that of the radiometer thermopile.

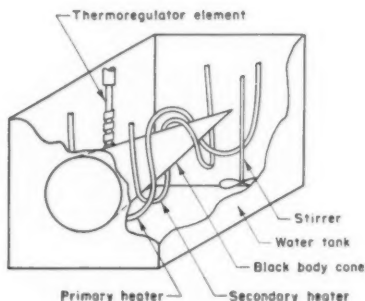


FIG. A-1. CUT AWAY VIEW OF BATH SHOWING BLACK BODY AND HEATERS

Several factors must be considered in determination of the radiometer constant and its subsequent use in calculation of radiation quantities from observations of the radiometer output. *First*, inequalities between the temperature of the thermopile element, the temperature of the tubular housing and the temperature of the shutter invariably existed. Hence, the instrument had a small output when the tube was closed off by the shutter. Since full response was reached in but a few seconds, the time elapsed during the sequence of the three readings referred to above was so short that changes in this *zero* output were either zero or so small that its change with time could be regarded as linear. As will be shown, this *zero* output could be eliminated from the output observed when the instrument sighted the black body. *Second*, the constant is dependent upon the mean temperature of the thermopile so that this dependence must be established. *Third*, vapor water in the line of sight absorbs a part of the radiation emitted by the object sighted and also contributes to the radiant energy which falls on the thermopile. The following equations express the relationship between radiometer outputs, its constant and the radiant exchanges between the thermopile and the objects which it sees:

1. Shutter open, radiometer sighted on black body:

$$\text{emf}_1/C = A_1 F_{1-2} \sigma [(1 - \epsilon_2) T_{b2}^4 + \epsilon_2 T_2^4 - T_1^4] + R \quad \dots \quad (\text{A-1})$$

## 2. Shutter closed:

$$\text{emf}_2/C = A_r F_{r-s} \sigma [(1 - \epsilon_1) T_{bl}^4 + \epsilon_1 T_1^4 - T_r^4] + R \quad \text{. . . (A-2)}$$

where

$\text{emf}$  = radiometer output in microvolts.

$C$  = radiometer constant,  $\mu V$  per Btu per hour which falls on the thermopile area.

$A_r$  = area of the radiometer thermopile, square feet (taken as the area of the diaphragm immediately in front of the pile).

$F_{r-s}$  = angle factor of the thermopile (the fraction of the radiation which passes through the outermost diaphragm that falls upon the thermopile).

$\sigma$  = Stephan-Boltzman constant =  $0.173 \times 10^{-8}$  Btu per (hour) (square foot) (Fahrenheit degree, absolute<sup>4</sup>).

$T_{bl}$  = temperature of the black body shutter, Fahrenheit absolute.

$T_{bs}$  = temperature of the black body, Fahrenheit absolute.

$T_r$  = temperature of the radiometer thermopile (jacket temperature), Fahrenheit absolute.

$T_1$  = average temperature of the water vapor column inside the radiometer tube, Fahrenheit absolute.

$\epsilon_1$  = emissivity of this water vapor column, dimensionless.

$T_2$  = average temperature of the water vapor column between the thermopile and object sighted, Fahrenheit absolute.

$\epsilon_2$  = emissivity of this water vapor column, dimensionless.

$R$  = net radiant energy exchange between the thermopile and its housing, Btu per hour.

Subtraction of Equation A-2 from Equation A-1 eliminates the unknown value of  $R$  and also the temperature of the thermopile itself, so that

$$[(\text{emf})_1 - (\text{emf})_2]/C = A_r F_{r-s} \sigma [(T_{bs}^4 - T_{bl}^4) - \epsilon_2 (T_{bs}^4 - T_2^4) - \epsilon_1 (T_1^4 - T_{bl}^4)] \quad \text{. . . (A-3)}$$

Ordinarily, temperatures  $T_2$  and  $T_1$ , the latter substantially equal to  $T_r$ , are nearly equal to  $T_{bl}$ , so that with negligible error Equation A-3 can be simplified as follows:

$$[(\text{emf})_1 - (\text{emf})_2]/C = A_r F_{r-s} \sigma [(T_{bs}^4 - T_{bl}^4) (1 - \epsilon_2)] \quad \text{. . . (A-4)}$$

Calibration over the temperature range 80 to 105 F and a range in  $\epsilon_2$  from 0.016 to 0.074 (dew points 22 to 78 F) established the constant at  $2.183 \times 10^4$  microvolts per Btu per hour radiation incident upon the thermopile area at 75 F. The correction for temperature was  $0.0018 \times 10^4$  increase in  $C$  per F deg decrease in radiometer temperature.

## APPENDIX B

### CALCULATION OF EMISSIVITY FROM EXPERIMENTAL DATA

The determination was similar to that followed in calibration, that is, the radiometer first saw the shutter at the end of the tube, black body No. 1, then the sample surface and once again black body No. 1. The outputs were recorded in each case. The output of the radiometer when it sighted the sample was due primarily to the net radiant energy exchange between the sample and the radiometer. However, it also included radiation from the surroundings as follows:

1. Radiation originating at the shield surface and being reflected from the sample surface into the radiometer.

2. Radiation originating at the sample surface and being reflected from the shield surface back to the sample surface and into the radiometer.

3. Radiation originating at the thermopile, a part of which is reflected from the sample back to the thermopile.

4. Radiation originating at either the shield surface or the sample surface and reaching the radiometer thermopile after multiple reflections.

The effect of item 4 is very small compared to the total reflected radiation and will not be considered. The contributions of items 1 and 2 are:

$$S_1 = A_r F_{r-s} \sigma (1 - \epsilon_s) [\epsilon_{sh}' (1 - \epsilon_s) T_{sh}^4] \quad \dots \quad (B-1)$$

$$S_2 = A_r F_{r-s} \sigma (1 - \epsilon_s) [\epsilon_s (1 - \epsilon_{sh}) (1 - \epsilon_s) (A_s/A_{sh}) T_s^4] \quad \dots \quad (B-2)$$

where

$\epsilon_{sh}$  = hemispherical emissivity of the cylindrical shield surface, dimensionless.  
 $\epsilon_{sh}'$  = effective emissivity of the cylindrical shield, dependent on  $\epsilon_{sh}$  and ratio of the shield to sample areas, dimensionless.

$\epsilon_s$  = emissivity of the sample, dimensionless.

$A_s$  = area of the sample, square feet.

$A_{sh}$  = area of the shield, square feet.

$T_s$  = temperature of the sample surface, F, absolute.

$T_{sh}$  = temperature of the shield surface, F, absolute.

The contribution of item 3 is

$$S_3 = A_r F_{r-s} \sigma T_r^4 (1 - \epsilon_s) (1 - \epsilon_s) M \quad \dots \quad (B-3)$$

where

$M$  = fraction of the reflected radiation that can be intercepted by the thermopile, dimensionless.

For diffuse reflecting surfaces,  $M$  is substantially zero in the apparatus described in this paper. For specular reflecting surfaces,  $M = 0.02$  approximately for normal incidence viewing and zero for all other angles. The direct contribution of the sample and the water vapor column is:

$$A_r F_{r-s} \sigma [\epsilon_s T_s^4 (1 - \epsilon_s) + \epsilon_2 T_2^4] \quad \dots \quad (B-4)$$

The radiometer develops an emf proportional to the net exchange between the thermopile and the objects it sees. When the instrument views the sample, this net exchange is:

$$\begin{aligned} \text{emf}/C &= A_r F_{r-s} \sigma [\epsilon_s T_s^4 (1 - \epsilon_s) + \epsilon_2 T_2^4] \\ &+ S_1 + S_2 + S_3 + R - A_r F_{r-s} \sigma T_r^4 \quad \dots \quad (B-5) \end{aligned}$$

By subtracting Equation A-2, which gives the emf when the shutter closes the end of the tube, the unknown  $R$  is eliminated. As before, in developing Equation A-4, the temperature of the water vapor column inside the tube and between the tube and the sample are taken equal to the temperature of the shutter. Solving for  $\epsilon_s$  gives the following equation:

$$\begin{aligned} \epsilon_s (T_s^4 - \epsilon_{sh}' T_{sh}^4) &= [(\text{emf})_1 - (\text{emf})_2] / [A_r F_{r-s} \sigma C (1 - \epsilon_s)] + T_{b1}^4 \\ &- [\epsilon_{sh}' T_{sh}^4 + 0.132 T_s^4 (1 - \epsilon_{sh}) (1 - \epsilon_s) \epsilon_s + T_r^4 (1 + Y)] \quad \dots \quad (B-6) \end{aligned}$$

where

$$\begin{aligned} Y &= M (1 - \epsilon_s) (1 - \epsilon_s) - 1 \\ A_s/A_{sh} &= 0.132 \text{ in the present apparatus.} \end{aligned}$$

Note that  $\epsilon_s$  appears in one of the quantities in the bracketed term in the numerator and also in  $Y$ . Equation B-6 is most easily evaluated by estimating  $\epsilon_s$  and then recalculating if necessary. Ordinarily, no further refinement is required.  $\epsilon_{sh}'$  was taken as unity for diffuse reflecting surfaces.

The specular reflecting surface presented a special problem in that only radiation from the shield that reached the sample at or very near the angle of viewing could reach the radiometer via a specular reflection. Hence, the effective emissivity of the shield was not necessarily unity. Moreover, the effective emissivity would be different

for normal emissivity than for all the other angles because of the sight tube.  $\epsilon_{sh}'$  was therefore experimentally determined by making tests with the shield at two different temperatures with the viewing angle,  $\theta$ , equal to zero in both cases. Since for angles close to zero incidence,  $\epsilon_s$  is the same, this factor was then used to find  $\epsilon_{sh}'$  for angles other than normal incidence by equating expressions for  $\epsilon_s$ . The effective shield emissivity,  $\epsilon_{sh}'$ , was found to be 0.882 for normal incidence and 0.90 for other than normal. The values of  $\epsilon_{sh}'$  determined included approximately the term  $0.132 T_s^4 (1 - \epsilon_s) (\epsilon_s)$ .

In some cases the sample may be so thick as to cause an appreciable temperature drop through it. This was true in the case of the asphalt, and therefore tests with two different thicknesses were made in order to estimate the temperature drop. This estimate checked very closely with direct measurement by thermocouples. Another way to minimize the possible error in determination of the temperature of the sample surface is to maintain the shield temperature at the temperature of the sample. The only heat loss to cause a temperature drop is that occasioned by the small radiation exchange between the sample and the radiometer. Temperature drop across the paint films was estimated by measuring the film thickness and calculating the temperature drop.

## DISCUSSION

H. C. BROWN, JR.,\* Lancaster, Penna., (WRITTEN): The authors of this paper are to be commended for the care taken in measuring the normal and hemispherical emissivity values. Without question the presentation of such data to our Society is of utmost value and is needed in modern engineering design particularly in panel heating and cooling. The data presented are of interest to the physicist because they confirm measurements for similar surfaces reported earlier by Hottel, McAdams and Jakob, and give further confirmation to the well known theoretical relationship between the normal and hemispherical emissivities.

Under Appendix A in Equation A-2, the question arises as to the application of the angle factor of the thermopile in determining the effect on the thermopile due to the emission by the water vapor column inside the radiometer tube. By definition the "angle factor" is only applicable to exchanges between the thermopile and surfaces outside the tube. Although the term  $\epsilon_s T_s^4$  is later dropped causing no effective error, it would still be interesting to know why this term is multiplied by the angle factor.

The use of a dimensionless "emissivity factor" to describe the effect of the water vapor column in absorbing and emitting radiation should be more fully explained. As this factor doubtless depends on the column length, it perhaps can be more accurately termed an "apparent emissivity" for the particular column length employed.

PAUL H. WEITZEL, Harrisburg, Penna.: I am interested in the texture effect on the angle function. May I ask the author this question before I proceed: What is the surface character of specular aluminum?

MR. PARMELEE: It is anodized aluminum.

MR. WEITZEL: Then I suppose the minute, elemental surfaces are not co-planar if analyzed under magnification.

MR. PARMELEE: I don't know.

MR. WEITZEL: My thought is that we are talking about: angles; emissivity of surfaces held at different angles; and surfaces which perhaps are not actually single planes as regards radiant behavior. I should like to suggest that the inherent irregularity of some types of surface has an effect on the angle function. Would you like to comment?

\* Armstrong Cork Company.

**AUTHORS' CLOSURE** (G. V. Parmelee): Mr. Brown questions the use of an emissivity factor with an angle factor to describe the radiation and absorbing characteristic of the column of water vapor in the line of sight of the thermopile. Equation A-2 is the radiant energy balance between the thermopile and its surroundings, first the material in the line of sight of the thermopile, and second the tube walls, diaphragms, etc. By definition the emissivity of a radiating gas is the ratio of the total radiant energy received from a hemispherical gas mass by an infinitesimal area or patch at the center of the base of this hemisphere to the radiation this area would receive from a "black" hemispherical surface of the same radius and at the same temperature as the gas with no enclosed absorbing medium. A hemisphere is chosen for the shape because the path length from the patch to the hemispherical boundary is the same in all directions. This definition also holds with respect to a conical gas shape of which the base can be imagined to lie beyond the outer diaphragm of the radiometer and to lie in the plane of and to be equal in area to that portion of some solid surface seen by the radiometer. The apex of the cone lies in the plane of the radiometer thermopile. Because the thermopile has a finite area, we actually were concerned with a truncated cone of gas and made the assumption that the emissivity (and absorptivity) of this gas shape was the same as that of the conical shape. The radiant energy exchange between the thermopile and the rest of the gas enclosed in the tube is included with the exchange between the pile and the radiometer tube. This has been designated by the term,  $R$ .

The emission of the conical gas column toward the thermopile is the emission of an imaginary "black" surface at the base of the cone times the emissivity,  $\epsilon_1$ , of the water vapor column. The "black" surface radiates toward the thermopile.

$$A_s F_{s \rightarrow r} \sigma T_1^4 \quad \text{Btu per hour}$$

where

- $A_s$  = area of the base of the cone, sq. ft.
- $F_{s \rightarrow r}$  = fraction of the total hemispherical radiation emitted by  $A_s$  through the outer diaphragm toward  $A_r$  the thermopile area.
- $T_1$  = temperature of the water vapor column, degrees F absolute.
- $\sigma$  = Stephan-Boltzman constant.

By the reciprocity theorem  $A_s F_{s \rightarrow r} = A_r F_{r \rightarrow s}$ , so that the second term of equation A-2 becomes  $\epsilon_1 A_r F_{r \rightarrow s} \sigma T_1^4$ .

This effect of water vapor on the instrument constant was verified by experiment. Calibrations and measurements were made at very low dew point and then with the highest possible dew point. Very satisfactory agreement was obtained for values of emissivity determined first at low dewpoint and then at high dew point when the corrections for water vapor absorption were applied.

In reply to Mr. Weitzel's question as to the nature of surfaces, the elemental surfaces of the specular aluminum were probably co-planar, at least for the wave length of light, because the material was a perfect reflector. It is not known if this is true for the long wave length infra-red. Undoubtedly the elemental surfaces of the other materials were not co-planar, because they were diffuse reflectors of light. It would seem that this lack of planeness should tend to minimize the departures from the cosine law at high angles of emission, rather than to cause the departures. The fact that the emissivity of the asphalt at high angles is actually greater than for the gray paint which has a higher normal emissivity and which presented a much smoother surface, may be evidence of this.



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## CIRCUIT ANALYSIS APPLIED TO LOAD ESTIMATING

### Part II—Influence of Transmitted Solar Radiation

By H. B. NOTTAGE\*, SANTA MONICA, CALIF., AND G. V. PARMELEE\*\*,  
CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

AIR-CONDITIONING engineers have been faced for many years with the problem of making suitable allowances in load estimates for the thermal capacity of the interior sections of buildings. Under conditions of periodic variations in solar intensity, outdoor temperature, and the outputs of various internal energy sources, these sections store and release in a cyclic manner part of the heat which enters an air-conditioned space. As a result, there can be a considerable difference between the instantaneous load on a cooling system and the calculated instantaneous rate of heat flow into the space. This has been demonstrated and pointed out in many papers.

Although practical means of calculating instantaneous rates of heat flow through weather-exposed building sections are in common use and take heat storage into account, means of adequately treating the heat-capacity effects of entire structures on cooling (or heating) system loads are still in the development stage. In the first ASHAE paper<sup>1</sup> on circuit analysis, the fundamental concept of representing thermal systems by thermal-circuit diagrams and a method of analyzing such circuits were developed. This concept recognizes that the thermal system of structure, equipment, furnishings, and outdoor environment constitutes a system of thermal capacitances, energy sources and energy receivers interconnected by thermal resistances. The thermal-circuit diagram shows these interconnections and thereby facilitates analysis. Moreover, the circuit diagram visually emphasizes the basic point that whatever happens at some instant of time at one point in the system or circuit influences the rest of the system.

This paper, and the first, are concerned mainly with developing principles of circuit diagramming and analysis. However, to sharpen reader familiarity with

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<sup>1</sup> Exponent numerals refer to References.

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the subject an illustrative problem has been set up and solved. For convenience, solution of the problem has been divided into:

*Phase A*—Analysis of cooling load due to the outdoor environment but *excluding* the effects of solar radiation transmitted directly through glass, and

*Phase B*—Analysis of cooling load due *only* to solar radiation transmitted through glass.

Results for Phase A only were given in the initial paper. The present paper gives the results for Phase B as (1), the load due only to transmitted-solar radiation and (2), this load plus the Phase A results.

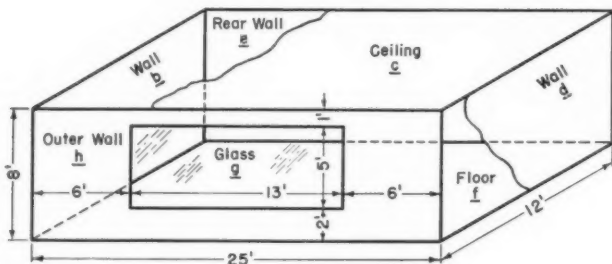


FIG. 1. SKETCH OF NORTH-EXPOSURE ROOM FOR ILLUSTRATIVE PROBLEM

Advisory guidance has been given by the Technical Advisory Committees on Cooling Load<sup>†</sup> and Heat Flow Through Glass<sup>‡</sup>.

#### ILLUSTRATIVE EXAMPLE

The thermal system established in the first paper and for this paper is the *north-exposure* room sketched in Fig. 1. The instantaneous load is to be found for an air-cooling system that maintains the room air at a temperature of 75 F throughout a 24-hr cycle.

Only loads arising from the influence of the outdoor environment are considered. Three energy sources are involved. One is the time-variable sol-air temperature for the weather side of the outer wall *h*. A second is the time-variable temperature of the window, which is warmed by the outdoor air and by absorption of part of the incident solar radiation. The load resulting from these two sources was given in the first paper<sup>1</sup> as Phase A results. The third load-producing source is the time-variable quantity of solar radiation which enters the room by direct trans-

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<sup>‡</sup> Personnel: R. A. Miller, Chairman; N. B. Hutcheon, Vice Chairman; W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, H. B. Vincent, C. J. Youngblood, Jr.

mission through the glass. In thermal-circuit terminology, the first two sources are described as *thermal-potential (temperature) generating* energy sources. The third is called a *thermal-current-generating* source.

As in Phase A, two cases are analyzed as follows: Case I—Single-glass window; Case II—Glass-block window. The glass block is that described in Chapter 13 of THE GUIDE 1954 as Type I.

TABLE 1—ASSUMED STRUCTURAL DETAILS AND THERMAL PROPERTIES FOR THE ILLUSTRATIVE EXAMPLE

ROOM COMPONENT	SYM-BOL, FIG. 1	MATERIAL	TOTAL THICK. IN.	SUR-FACE AREA SQ FT	$\frac{k}{\text{BTU}/(\text{HR}) (\text{SQ FT}) (\text{F DEG}/(\text{FT}))}$	$\rho$ LB	$c_p$ BTU	$a$ SQ FT
						CU FT	LB F DEG	HR
Floor	<i>f</i>	Concrete, no covering	4	300	0.41	97	0.21	0.020
Ceiling	<i>c</i>	Same as floor						
Interior Partitions	<i>e</i>	Gypsum lath & plaster	$\frac{7}{8}$	200	0.275	70	0.23	0.017
		Wood studs, $1\frac{3}{4}$ x $3\frac{3}{4}$ in. spaced 16-in centers	—	—	0.085	30	0.40	0.007
	<i>b, d</i>	Same		96				
Exterior Wall	<i>h</i>	Brick	4	135	0.51 <sup>a</sup>	112 <sup>a</sup>	0.20	0.023
		Mortar	$\frac{1}{2}$	135	1.0	115	0.22	0.039
		Cinder-concrete blocks	8	135	0.40 <sup>a</sup>	57 <sup>a</sup>	0.21	0.033
		Furring space, lath & plaster, taken to be same as for interior partitions.						
Glass	<i>g</i>	Case I—Single common window glass—65 sq ft Case II—Type I 8-in. glass blocks Smooth exterior faces, wide ribs or flutes on interior faces—65 sq ft						

<sup>a</sup> For solid substance with no core holes, having the same net weight and overall thermal resistance as the actual structural units. Symbols:  $k$  = thermal conductivity;  $\rho$  = density;  $c_p$  = unit heat capacity at constant pressure;  $a$  = thermal diffusivity.

Structural details are the same as for Phase A, but are repeated in this paper for convenience in Table 1. The postulated conditions which further define the problem include the condition of no heat flow across the mid-planes of the ceiling, the floor and the interior walls. In effect, then, the adjoining spaces are at the same temperature as the room chosen for analysis.

In order to approach a practical situation, the weather conditions chosen for this *north-exposure* room are those for a clear atmosphere on an August 1 design day at 40 deg North latitude. The diurnal cycle of dry-bulb temperature is that given in Table 8 of Chapter 13 of THE GUIDE 1954 and has a maximum of 95 F at 3:00 p.m. suntime and a minimum of 74 F at 5:00 a.m. The solar data used in the problem are discussed later.

## CIRCUIT DIAGRAM

The thermal circuit which represents this room under the imposed conditions is Fig. 2. The lettered junction points correspond to the lettered room surfaces of Fig. 1. The constant room-air temperature is junction point  $a$  in the center of the diagram. The dashed branches of the circuit connect the directly transmitted solar radiation, which enters the room through the glass, with the various room surfaces. Thermal-capacitance elements are designated as  $C_k$ ,  $C_m$ ,  $C_f$ , etc. and thermal-resistance elements as  $R_{mh}$ ,  $R_{ha}$ ,  $R_{hf}$ , etc. The thermal capacitance of the outer wall is the sum of the first two designated capacitances. The third equals half of the thermal capacitance of the floor (the other half would appear in the circuit for the adjoining room). The three designated resistances are respectively: part of the thermal resistance of the outer wall  $h$ ; the convection resistance between  $h$  and the room air  $a$ ; and the radiation resistance between wall surface  $h$  and floor surface  $f$ . Evaluations of these circuit quantities were described in appendices to the first paper. The values are summarized in Appendix A of this paper.

## CIRCUIT PRINCIPLES

At this point, a brief explanation of circuit principles may encourage the reader to look upon the circuit diagram as a vivid *picture* of a typical *thermal system*. The circuit should *not* be regarded as an electrical circuit although electrical symbols were borrowed to represent thermal elements and although analogous electrical circuits are a useful means of solving thermal problems.

The basis for drawing and solving thermal circuits is the *principle of the thermal-current or heat balance*. Consider point  $f$  on the floor surface. A solar-radiation thermal current comes through the window and strikes the floor where it is absorbed. In the exchange of low-temperature radiation between  $f$  and the other room surfaces, thermal currents flow between these surfaces and  $f$  through radiation resistances, such as  $R_{hf}$ , connecting these surfaces with  $f$ . At the same time a convective thermal current flows from  $f$  to room air  $a$  (this is a cooling load current) through resistance  $R_{fa}$ . Also a conductive thermal current flows from  $f$  through resistance  $R_f$  into capacitance  $C_f$ , which stores this energy. At any instant of time, the algebraic sum of these thermal currents must equal zero in accord with the law of the conservation of energy.

Solving circuit problems consists simply in expressing the thermal-current balances for each junction point in mathematical equations and solving them simultaneously for the unknown circuit-junction temperatures. Thermal currents can then be computed. Alternately, some sort of physical analogue can be constructed and measurements made by instruments, a procedure similar to studying the performance of a full-scale thermal system. Further discussion will be confined to the mathematical method and other techniques developed in this and the previous paper.<sup>1</sup>

*Superposition:* In this paper, the thermal circuit is treated as a *linear* circuit; that is, one in which the thermal properties of the materials and the convection and radiation resistances are independent of temperature and rates of heat flow. Such a circuit permits direct *superposition* of the effects of all inputs treated separately and is the basis for analyzing the circuit in Phases A and B. For example, if, at some instant of time, the temperature of the floor surface  $f$  is found to be 1.5 deg above the constant room-air temperature of 75 F under the Phase A

conditions only, and 1.2 deg above 75 F under the Phase B conditions only, the floor temperature for the combined conditions would be 75 F plus the sum of 1.5 and 1.2 or 77.7 F. Thermal currents are similarly added.

Though linearization of circuits is an idealization from the real system, a very important advantage is gained, namely, the ability to determine the separate contribution of each load-producing element to the total system load. Furthermore, computation procedures (or analogue techniques) in circuit analysis are simplified.

In the Phase A calculations, the sol-air and glass inner-surface temperatures were established by calculations based upon the chosen solar-radiation values, air

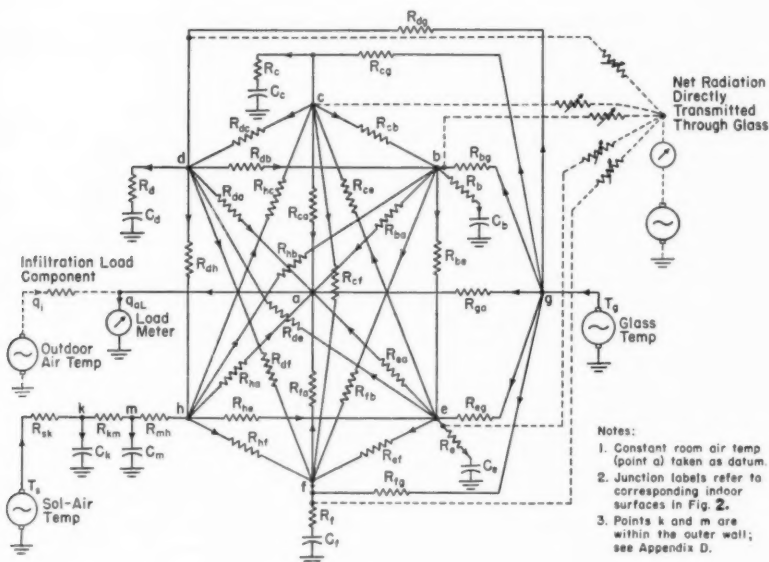


FIG. 2. THERMAL CIRCUIT TO BE SOLVED

temperature and outdoor convection conductance. The junction-point temperatures and thermal currents resulting from only these two cyclic potentials were solved for in Phase A. The solar-radiation current generator was removed and the dotted branch connections were left open.

In the Phase B problem, the circuit is solved under the condition that the transmitted-solar-radiation thermal-current source is connected via the dotted lines while the sol-air and glass-temperature generators are removed, leaving g and s connected to ground through the dotted jumpers shown in Fig. 2. This treatment is not essential to solving the circuit, but permits the solar-radiation load currents to be determined independently of the currents produced by other sources. This step is validated in Appendix B.

One more point with respect to circuit handling technique requires mention, namely, the matter of not connecting  $g$  to ground through an outdoor convection resistance. This is discussed in Appendix C.

**Transmitted-Solar-Radiation Thermal Currents:** The total transmitted-solar-radiation thermal current is the sum of a diffuse (or sky) component and a direct (or beamed) component. The thermal currents for this problem are shown in Fig. 3. With respect to the diffuse component, it was assumed that partial shading by the window opening was negligible. Hence hourly values of this component were multiplied by the window opening area of 65 sq ft. On the August 1 design day chosen for this problem, the direct component strikes a north-exposure wall

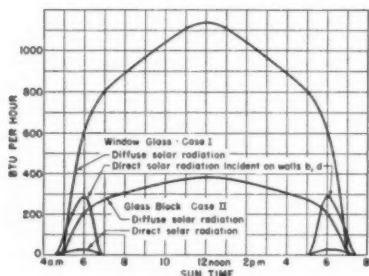


FIG. 3. TRANSMITTED SOLAR RADIATION THERMAL CURRENTS THROUGH WINDOW GLASS AND GLASS BLOCK TYPE I FOR ILLUSTRATIVE PROBLEM

or window only for a few hours early in the morning and again late in the afternoon. Moreover, the beam strikes the opening at such large incident angles that most of it is intercepted by the sides of the window opening in the thick wall. Therefore, hourly values of the direct component were multiplied by 65 sq ft times the fractional cross-section area of the beam which actually enters the room. These fractions are given in Table D-1 of Appendix D.

Hourly values of the diffuse and direct components which were used in computing the thermal currents of Fig. 3 were those used by the ASHAE Research Laboratory in preparing heat gain data tables for THE GUIDE. The sums of these components now appear in Tables 13 and 20 of Chapter 13 of THE GUIDE 1954 as instantaneous rates of heat gain due to transmitted direct and diffuse solar radiation by single common window glass and Type I glass block respectively. However, because the table values are rounded off to the nearest whole number, there are small differences between the table values times 65 sq ft and the curves of Fig. 3. For example, at noon the table value of 17 Btu per (hr) (sq ft) for north-exposed window glass on August 1 at 40 deg north latitude gives a total of 1105 Btu per hr whereas the diffuse component curve of Fig. 3 gives 1140, a value which also appears later on in the fourth column of Table 3. The result of partial shading of the direct beam is brought out in the following example: At 6:00 a.m. (and 6:00 p.m.) the table value of 26 Btu per (hr) (sq ft) given in THE GUIDE gives

a total of 1690 Btu per hr for window glass, whereas in this paper the sum of the direct and diffuse components is  $(16.5 \times 0.268 + 9.5)$  65 or 905 Btu per hr.

The distribution of the thermal currents among the room surfaces was determined as follows. With respect to the diffuse component it was assumed that the transmitted radiation emanated uniformly from the entire glass surface. The total, therefore, was divided among the various room surfaces in accordance with their shape factors with respect to the window. These fractions are given in Table D-1 of Appendix D and are the same for both Case I and Case II. This division is indicated by the dotted lines to the several junction points from the transmitted-radiation thermal-current source (Fig. 2). It was further assumed that the room surfaces completely absorbed the solar radiation incident upon them.

A geometrical analysis of the direct beam shows that, of the beam *not* cut off by the window opening, only west wall *d* intercepts the beam in the morning and

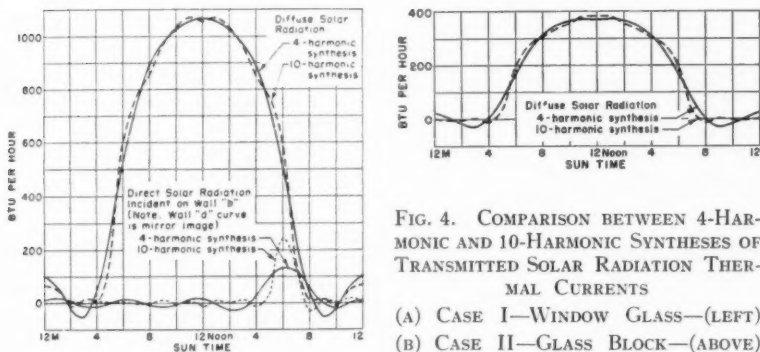


FIG. 4. COMPARISON BETWEEN 4-HARMONIC AND 10-HARMONIC SYNTHESES OF TRANSMITTED SOLAR RADIATION THERMAL CURRENTS

(A) CASE I—WINDOW GLASS—(LEFT)  
(B) CASE II—GLASS BLOCK—(ABOVE)

only east wall *b* in the afternoon. To avoid unnecessary complications in this illustrative problem, it was assumed that the plane of the glass coincides with the plane of the inner surface of wall *h*. Hence, the portion of the beam intercepted by the sides of the opening falls outside the room. It was assumed, therefore, that the energy in this part of the beam was dissipated to the outdoors. Note that the dotted connections to *b* and *d* carry diffuse radiation thermal currents and, at certain times, they carry these direct radiation currents as well.

A further assumption with respect to these two beams was that surfaces *b* and *d* maintain uniform temperatures at all times. Were this not done, the analysis would require each of these surfaces to be divided into a beam intercepting and a shadowed portion and the introduction of additional junction points in the circuit diagram. (This latter step might well be justified in some problems.)

In Case II, the transmitted direct-radiation currents are so small that they have been added directly to the load to obtain the total resultant loads. It should be emphasized that spatial distribution data on radiation from sources such as the sun and lighting fixtures are required in order to supply the proper current values to each room surface.

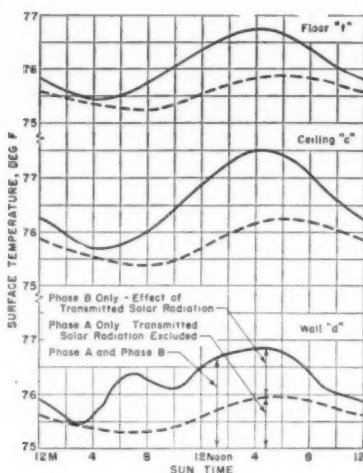


FIG. 5. DIURNAL VARIATIONS OF TEMPERATURES OF SELECTED ROOM SURFACES AS AFFECTED BY CONDITIONS OF PHASES A AND B COMPARED WITH EFFECT OF CONDITIONS OF PHASE A ONLY: CASE I—WINDOW GLASS

*Fourier Series Expressions:* The technique of circuit analysis used in this and the previous paper requires that periodic phenomena, such as sol-air temperatures and transmitted radiation currents, be expressed as Fourier series. Therefore, analyses of the curves of Fig. 3 were made through the first 10 harmonics. Table D-2 of Appendix D summarizes the harmonic coefficients. Figs. 4a and 4b give the four-harmonic and ten-harmonic syntheses. Comparison shows that the four-harmonic syntheses closely approximate the corresponding curves of Fig. 3 and therefore were used in the Phase B calculations. This point is discussed further at the end of the paper.

*Circuit Equations and Their Solution:* Circuit equations are written by making instantaneous thermal current balances on each junction point in Fig. 2. As previously noted the instantaneous algebraic sum of all the currents entering a junction point must be zero. In thermal circuits current flow in both resistance and capacitance branches must be considered. This is treated by the concepts of thermal admittance and thermal impedance, which were explained in the first paper<sup>1</sup>. Admittance and impedance are analogous to conductance and resistance respectively, so that

$$\text{Thermal Current} = \frac{\text{Thermal Potential Difference}}{\text{Thermal Impedance}} \dots (1)$$

$$\text{Thermal Current} = \text{Thermal Admittance} \times \text{Thermal Potential Difference} \dots (2)$$

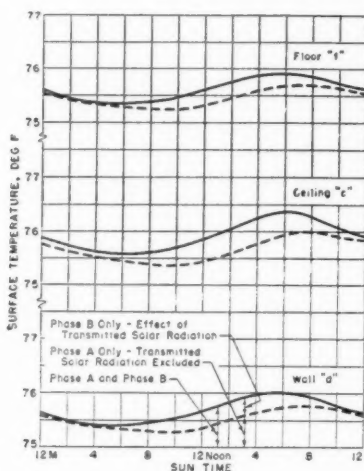


FIG. 6. DIURNAL VARIATIONS OF TEMPERATURES OF SELECTED ROOM SURFACES AS AFFECTED BY CONDITIONS OF PHASES A AND B COMPARED WITH EFFECT OF CONDITIONS OF PHASE A ONLY:  
CASE II—GLASS BLOCK

where

Thermal Current	= Btu per hour
Thermal Potential Difference	= Fahrenheit degrees
Thermal Impedance	= Fahrenheit degrees per (Btu) (hour)
Thermal Admittance	= Btu per hour (Fahrenheit degree)

The various currents, then, are expressed in terms of junction potentials and branch admittances in accord with Equation 2, or as quantities, such as the solar radiation thermal currents, which are known from the conditions of the problem. The number of equations is equal to the number of unknown junction potentials. The techniques of writing equations and the use of *vector notation* have been explained in the previous paper.

Tables D-3 and D-4 of Appendix D give the equations to be solved. Solutions were obtained by means of a computing machine designed to solve simultaneous linear algebraic equations and are in the form of Fourier series coefficients for the junction-point potentials. These are listed in Table D-5.

## RESULTS AND DISCUSSION

*Structural Temperature Cycles:* Figs. 5 and 6 give the temperature cycles for the floor, ceiling, and end wall *d*, and show the influence of directly transmitted radia-

tion by comparison of the Phase A and Phase B magnitudes. Curves for the other surfaces can be plotted from the data of Table D-5.

The bump in the curve for wall *d* in Case I is the result of direct or beamed radiation falling on this surface for about two hours early in the morning, in addition

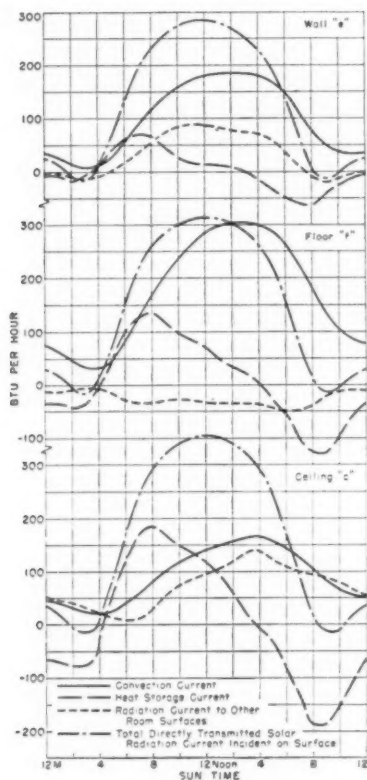


FIG. 7. THERMAL CURRENTS FOR CASE I—WINDOW GLASS—PHASE B ONLY

to the diffuse radiation, which increases slowly to a maximum at noon and then decreases. The differences between the pairs of curves represent the maximum attainable improvement by shading devices. It is worth noting that though the room-air temperature is held constant, the area-weighted average temperature of the room surfaces is somewhat higher. In Case I this temperature was a maximum of 2.8 deg above the room air temperature for Phases A and B combined.

Note further that orientations other than the north used for this problem receive much more solar radiation.

*Structural Thermal-Current Cycles:* Currents in the various circuit branches are computed as the vector products of the corresponding admittances and potential

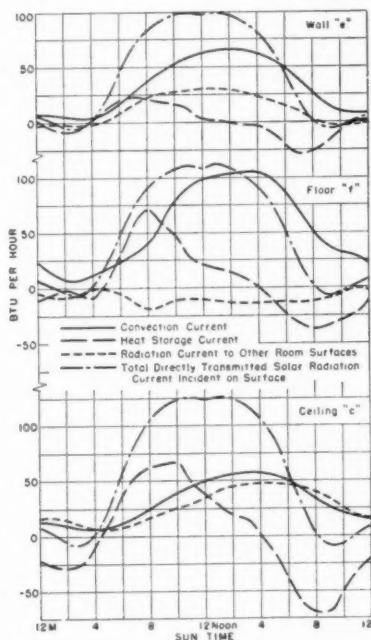


FIG. 8. THERMAL CURRENTS FOR CASE II—GLASS BLOCK—PHASE B ONLY

differences. Complete current cycles for the floor, ceiling, and wall *e* (opposite the glass) are given in Fig. 7 for Case I, single window glass, and in Fig. 8, Case II, glass block. Resultant currents are given in Fig. 9 and show the effect of thermal storage on load due to directly transmitted radiation. For Case I in Fig. 9 the solar radiation curve is the sum of the four-harmonic synthesized curve of transmitted diffuse solar radiation and the synthesized curves of transmitted direct radiation which falls on end walls *b* and *d*. These are shown separately in Fig. 4a. For Case II the curve is identical with the four-harmonic synthesized curve of Fig. 4b. The small direct-to-load radiation component of Case II is not included.

The cycles of individual current balances (heat balances) largely tell their own story of the interplay of heat-transfer and heat-storage phenomena. Note particularly the following five points.

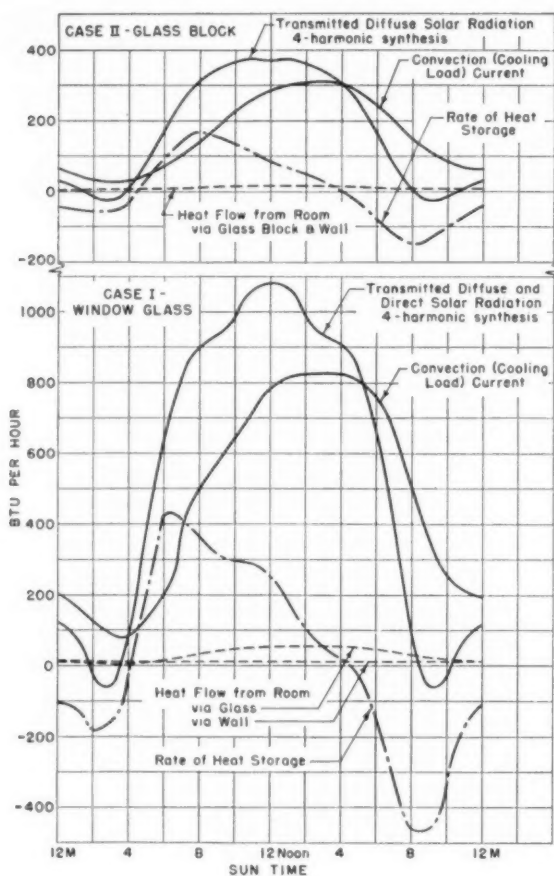


FIG. 9. RESULTANT CONVECTION (COOLING LOAD), HEAT STORAGE AND HEAT-FLOW-FROM-ROOM CURRENTS COMPARED WITH TRANSMITTED SOLAR RADIATION FOR PHASE B ONLY—NOTE: DIRECT-TO-LOAD SOLAR RADIATION CURRENTS ARE EXCLUDED

1. The resultant convection currents (system load) due to transmitted solar radiation of Fig. 9 are of especial interest. In both cases they lag the peaks of the solar-radiation currents by approximately three hours and the peak convection currents are 75 and 80 percent of the peak radiation currents for Cases I and II.

2. Note that even at midnight the system load is 20 to 25 percent of the peak load. This, of course, is due to the release of stored heat, indicated by the negative rates of

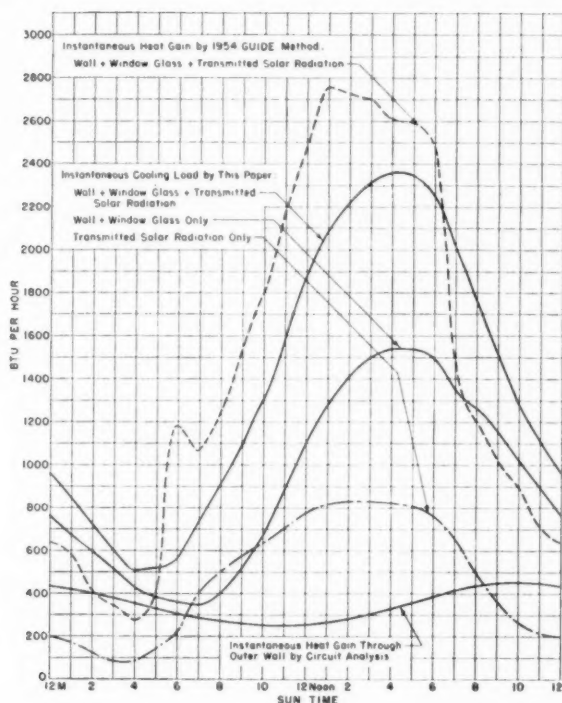


FIG. 10. INSTANTANEOUS COOLING LOADS COMPARED WITH INSTANTANEOUS HEAT GAINS: CASE I—WINDOW GLASS

heat storage shown in Fig. 9 between the hours of 4:00 p.m. and 4:00 a.m. in both Case I and Case II.

3. Figs. 7 and 8 show that the low temperature radiation exchange is from the other room surfaces to the floor, that is, the floor is the coolest surface. The floor absorbs nearly as much radiation per unit area as the other room surfaces, but its higher convection conductance prevents it from reaching temperatures as high as those of the others.

4. The net heat balance on all interior surfaces gives no net flow through these surfaces throughout a 24-hr period. However, in this period there is a small flow from the room to the outdoor environment through the glass and through the exposed wall  $h$ . Had phases A and B been solved together, this current flow would have appeared as a reduction in the heat conduction through the glass from the warmer outdoors to the room. For Case I, separate curves have been shown. Because of the low thermal resistance, the heat flow out through the glass is nearly 8 percent of the total input. (This is discussed in Appendix C.) Because of the high thermal resistance of the wall and the glass block and the smaller transmitted radiation, this flow is very small in Case II. The sum is represented by a single curve in this instance.

5. The reduced transmission from glass block shows up strongly.

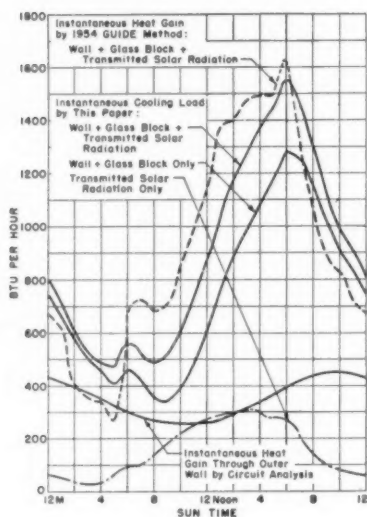


FIG. 11. INSTANTANEOUS COOLING LOADS COMPARED WITH INSTANTANEOUS HEAT GAINS: CASE II—GLASS BLOCK

**Total System Load:** Figs. 10 and 11 give the total resultant cooling loads by combining the Phase A results of the previous paper and the load curves of Fig. 9. The small direct-to-load component of Case II is also included in Fig. 11. The values are also listed in Table 2. In both cases the load peak of Phases A and B together occurs at the same hour as the Phase A load peak despite the fact that the Phase B results peak about 2 hr earlier. This is because the Phase A conditions are the principal load producing factors for this north orientated room. Though the Phase A results do not separate the load due to conduction through the wall from the load due to the glass being warmed by both the sun and outdoor air, the warm glass surface appears to be the dominant contributor to the total cooling load.

**Instantaneous Cooling Load and Instantaneous Heat Gain:** In the absence of methods of treating internal heat storage, common practice is to consider the instantaneous rates of heat gain, such as those given in THE GUIDE, as instantaneous cooling loads. Hence, the synthesized radiation curves of Fig. 9 represent cooling load by this standard. In this particular example, the instantaneous gain method overestimates the load due to the transmitted radiation component and gives an earlier time for the peak load.

Recently a similar circuit was analyzed by Mackey and Gay<sup>2</sup> with a hydraulic analogue. With comparable ratio of glass to floor area but with a *south* exposure the peak cooling load was found to be about 50 percent of the peak instantaneous rate of heat gain and occurred about 2 hr later. This compares with 75 to 80 percent and a 3-hr time lag in the present paper for the north-exposure room. The reason for the differences, particularly the relative amplitudes, may be due to the more rapid rate of increase in the south wall solar radiation curve.

Figs. 10 and 11 compare the total instantaneous cooling loads with the total instantaneous heat gains. These values are listed in Table 3. In Case I the peak load occurs 3 hr later than the peak gain and is 15 percent less. In Case II the result is different in that the peaks occur at the same time and the total peak load is only 5 percent less than the peak instantaneous gain.

TABLE 2—INSTANTANEOUS SYSTEM-LOAD COMPONENTS BY CIRCUIT ANALYSIS

HEAT-FLOW RATES—BTU PER HOUR							
SUN TIME	CASE I—WINDOW GLASS			CASE II—GLASS BLOCK			
	PHASE A	PHASE <sup>a</sup> B	TOTAL	PHASE A	PHASE B		TOTAL
					DIFFUSE	DIRECT <sup>b</sup>	
12 Mn	763	204	967	747	62		809
1 a.m.	673	173	846	667	52		719
2	596	126	722	576	33		609
3	515	87	602	506	25		531
4	427	84	511	460	29		489
5	382	145	527	410	44	14	468
6	361	205	566	460	69	28	557
7	347	398	745	421	103		524
8	398	500	898	353	140		493
9	513	575	1088	342	184		526
10	666	643	1309	388	224		612
11	883	713	1596	479	258		737
12 N	1101	783	1884	608	280		888
1 p.m.	1276	818	2094	751	296		1047
2	1406	824	2230	886	302		1188
3	1499	824	2323	985	306		1291
4	1539	823	2362	1079	301		1380
5	1540	806	2346	1174	279		1453
6	1501	757	2258	1280	244	28	1552
7	1348	657	2005	1256	199	14	1469
8	1264	500	1764	1154	145		1299
9	1154	369	1523	1023	105		1128
10	1018	264	1282	909	83		992
11	892	219	1111	838	73		911

<sup>a</sup> Due to both diffuse and direct solar radiation.

<sup>b</sup> This is the *Direct-to-Load* component.

The total instantaneous gains listed in Table 3 and plotted in Figs. 10 and 11 were prepared as follows. The heat flows through the exposed north wall as given in Table 3 were computed in accordance with Equation 5 of Chapter 12 of THE GUIDE 1952. The sol-air temperatures used were those given by Fig. 4 of Reference 1. Convection and radiation gain values for single glass and for glass block were taken directly from Tables 17 and 24, respectively, of Chapter 12 of THE GUIDE 1952 and multiplied by 65 sq ft to obtain the C & R values listed in Table 3. Note that these tables are based on 75 F indoor temperature, the same condition postulated for this illustrative problem. (The 1952 values were used in order to be consistent with the first paper, which was prepared in 1953 when only THE GUIDE

1952 was available.) The heat flow rates due to transmitted solar radiation are simply the sums of the direct and diffuse components given by the curves of Fig. 3. The preparation of these data was discussed earlier in this paper. The total instantaneous gains shown in Figs. 10 and 11 are the sums of these three heat gain components.

TABLE 3—INSTANTANEOUS HEAT-FLOW RATES FOR ILLUSTRATIVE PROBLEM

HEAT-FLOW RATES—BTU PER HOUR							
SUN TIME	OUTER WALL TOTAL	SINGLE GLASS C & R	SINGLE GLASS TRANSMITTED	WALL & SINGLE GLASS TOTAL	GLASS BLOCK C & R	GLASS BLOCK TRANSMITTED	WALL & GLASS BLOCK TOTAL
12 Mm	520	117	0	637	156	0	676
1 a.m.	531	52	0	583	91	0	622
2	377	33	0	410	33	0	410
3	345	0	0	345	13	0	358
4	321	-46	0	275	13	0	334
5	301	-65	131	367	-65	27	263
6	281	0	905	1186	130	236	647
7	257	0	806	1063	195	273	725
8	246	130	891	1267	130	306	682
9	246	325	969	1540	130	332	708
10	235	520	1040	1795	260	358	853
11	224	845	1110	2179	390	377	991
12 N	235	1105	1140	2480	520	384	1139
1 p.m.	345	1300	1110	2755	650	377	1372
2	321	1365	1040	2726	715	358	1394
3	301	1430	969	2700	845	332	1478
4	345	1365	891	2610	845	306	1496
5	377	1300	806	2583	845	273	1495
6	421	1170	905	2496	975	236	1632
7	465	845	131	1441	845	27	1337
8	487	715	0	1202	585	0	1072
9	498	520	0	1018	390	0	888
10	509	384	0	893	320	0	829
11	487	220	0	707	234	0	721

## GENERAL COMMENTS

*Possible Correction Factors to Instantaneous Heat Gain:* The present results are limited, but it appears promising to take the instantaneous load as some percentage of the instantaneous gain as computed from data in THE GUIDE. However, because the Phase B results alone give substantially the same time lag and nearly the same ratio of peak load to peak gain for both cases, whereas this is not true when the Phase A and Phase B results are considered together, it appears that percentages should be applied to load components. A method of adjusting instantaneous rates of heat gain in residences for heat storage, based on analogue computer studies, is given in the recent paper by Willcox, et al<sup>3</sup>. It is apparent that much more study of typical circuits and particularly of the separate effects of load-producing components is needed. With respect to the effects of internal heat storage on loads, one can expect that spaces cooled by circulated air alone will differ considerably from spaces cooled by air and cool panels. The present problem indicates the importance of using shading devices, but the fact that interior shades

can reach rather high temperatures and convert much of the radiation component to convection must be taken into account.

Though circuit analysis indicates that significant reduction factors on instantaneous heat gain rates with consequent reduction in size of equipment are in order, further reduction in size of equipment can also be made by allowing the indoor temperature to rise at the peak. Circuit analysis principles provide an excellent tool to study this and other operating procedures.

*Infiltration:* Note that infiltration is considered to be an instantaneous load, added directly to the system load in Fig. 2. Within the uncertainties or infiltration, this is judged to be reasonable.

*Fourier Series Treatment:* The burden of detailed calculations in the solution of circuit problems by the equation method, as employed in this study, arises from the need to consider several Fourier-series harmonics. Present experience suggests that no more than four harmonics will be needed in practical problems. Judgment rests upon the shapes of the input cycles,—the nearer these are to sine waves, the fewer the needed harmonics.

The accuracy to which a harmonic analysis or synthesis *should* be carried depends upon the purpose involved. Detailed study of temperature and heat-flow cycles requires more harmonics than does a gross study of system load. Referring to Fig. 4, the gain from four to ten harmonics is seen not to be worth the effort in the present case. The four-harmonic synthesized input curve of diffuse radiation, for Case I, for example, makes the sun rise at about 3:30 a.m. whereas the correct time of sunrise is 4:30 a.m. (see Fig. 3). Moreover, it introduces a radiation current during the night hours that oscillates between a positive 100 Btu per hour and a negative 55 Btu per hr. Additional harmonics improve the representation of the true cycle, but for the north exposure, the departures from the behavior of a real physical system are not serious.

It should be noted that the addition of harmonics to the 24-hr average or steady-state value changes only the shape of the curve but does not change the net area under the curve. Furthermore, the net area under a curve synthesized by adding any number of harmonics equals the area under the curve which has been "analyzed". That is, the areas under the synthesized diffuse solar radiation curves of Fig. 4a are exactly equal to the area under the diffuse solar radiation curve for window glass given in Fig. 3.

It is probable that all cases where such discontinuous functions are involved will require a considerable number of harmonics for proper representation. It should be noted that the radiation input from lights which are on only part of the day would be a discontinuous or step-wave input function. In contrast to this, harmonic representations of sol-air temperatures require relatively few harmonics because such functions oscillate continuously throughout the 24-hour cycle.

More experience will give a basis for generalizing and simplifying the series analysis.

#### FURTHER WORK

The really lasting value of the present study is the further demonstration and validation of the circuit method for continuing work in load analyses, control studies, and all types of structural and equipment problems wherein the *system* effect of any variable is needed. This viewpoint is very important for economic analyses, wherein the importance of a particular quantity (resistance, capacitance, potential, or current) is related to the expenditure connected with a change or

control of its cyclic magnitude. The system problems of automatic controls, in particular, can be properly studied only with some form of circuit analysis.

The next step needed is an analysis of the same room with different exposures, and then of the same room but located directly under a roof. Shading devices also should be studied.

It is entirely possible that sufficient experience in the load-estimating use of circuit analysis will lead to a simple and practical classification of structures, with factors which will correct the load estimating method as given in THE GUIDE to an actual equipment-selection basis. This thought also enters via the work of others<sup>2, 3</sup>.

Complicated systems, wherein the value of the results warrants the cost, perhaps are best handled on an *analogue*. The ASHAE has taken steps in this direction through sponsorship of a cooperative project at the University of California, Los Angeles, and the appointment of a Coordinating Committee on Thermal Circuit to advise concerning an analogue at the ASHAE Research Laboratory.

### CONCLUSIONS

1. The practical value of circuit methods for analyzing air-conditioning loads has been further demonstrated in an illustrative example.

2. For a north-exposed room, the peak instantaneous cooling load due only to solar radiation transmitted through a window was found to be 75 percent of the peak instantaneous heat gain and to occur 3 hr later.

3. Continuing work is essential both to realize the full potentialities of circuit analyses and to simplify the solution of problems.

### REFERENCES

1. ASHVE Research Report No. 1497—Circuit Analysis Applied to Load Estimating, by H. B. Nottage and G. V. Parmelee (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 59).
2. Cooling Loads from Sunlit Glass and Wall, by C. O. Mackey and N. R. Gay (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 469).
3. Analogue Computer Analysis of Residential Cooling Loads, by T. N. Willcox, C. T. Oergel, S. G. Reque, C. M. toeLaer, and W. R. Briskin (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 505).

### APPENDIX A

TABLE A-1—SUMMARY OF CIRCUIT RESISTANCES AND CAPACITANCES

Branch Resistances  $\times 10^2$  in F deg per (Btu)(hr)  
Example:  $R_{de} = 0.0575$  F deg per (Btu)(hr)

BRANCH	b	d	h	e	c	f	m	k	g	s
c	3.97	3.97	2.82	1.80	—	0.857	—	—	5.02	—
d	19.4	—	7.14	5.75	3.97	3.97	—	—	29.58	—
b	—	19.4	7.14	5.75	3.97	3.97	—	—	29.58	—
e	5.75	5.75	4.14	—	1.80	1.80	—	—	6.44	—
f	3.97	3.97	2.63	1.80	0.857	—	—	—	5.73	—
h	7.32	7.32	—	4.14	2.82	2.63	1.03	—	—	—
m	—	—	1.03	—	—	—	—	1.37	—	—
k	—	—	—	—	—	—	1.37	—	—	0.350
a	1.49	1.49	1.06	0.714	0.834	0.303	—	—	2.20	—
To Ground	0.167	0.167	—	0.080	0.068	0.068	—	—	—	—

TABLE A-1 (CONTINUED)—SUMMARY OF CIRCUIT RESISTANCES AND CAPACITANCES

Capacitances and Reactances

Example:  $X_{cb}$  for  $f = (1/24) \text{ hr}^{-1} = 0.0289$ 

BRANCH	<i>b</i>	<i>d</i>	<i>h</i>	<i>e</i>	<i>c</i>	<i>f</i>	<i>m</i>	<i>k</i>
Capacitance, <i>C</i> , Btu/F								
	132.	132.		276.	1020.	1020.	1260.	1150.
Capacitance Reactance $\times 10^2$ , $X_c^a$								
$f = 1/24 \text{ hr}^{-1}$	2.89	2.89		1.385	0.375	0.375	0.303	0.332
$f = 1/12$	1.45	1.45		0.693	0.187	0.187	0.152	0.166
$f = 1/8$	0.965	0.965		0.461	0.125	0.125	0.101	0.111
$f = 1/6$	0.724	0.724		0.346	0.937	0.937	0.0758	0.0831

<sup>a</sup>  $X_c = \frac{1}{2\pi fC}$ , where  $f$  = frequency.

## APPENDIX B

### SUPERPOSITION IN LINEAR CIRCUITS

The procedure of removing the sol-air temperature and glass temperature *generators* from the circuit and connecting the respective branches directly to ground warrants further discussion. Note that it is permissible only if the circuit is *linear*. The proof of this step is as follows:

Consider point *k* of Fig. 2 and let the temperatures for Phase A be designated as  $T_k$ ,  $T_m$ , etc., and for Phase A plus Phase B as  $T'_k$ ,  $T'_m$ , etc. Ground potential in each case is the same,  $T_o$ . Recall from equation 1 that the current in any branch is a product of the admittance of the branch,  $Y$ , times the temperature difference. The equations for point *k* are:

#### PHASE A

$$(T_o - T_k)Y_{sk} = (T_k - T_o)Y_{ck} + (T_k - T_m)Y_{km} \dots \quad (\text{B-1})$$

#### PHASE A AND PHASE B

$$(T_o - T'_k)Y_{sk} = (T'_k - T_o)Y_{ck} + (T'_k - T'_m)Y_{km} \dots \quad (\text{B-2})$$

For convenience,  $T_o$  is usually taken as zero potential; thus with terms collected, the equations become respectively:

$$T_o Y_{sk} = T_k(Y_{ck} + Y_{km} + Y_{sk}) - T_m Y_{km} \dots \quad (\text{B-3})$$

$$T_o Y_{sk} = T'_k(Y_{ck} + Y_{km} + Y_{sk}) - T'_m Y_{km} \dots \quad (\text{B-4})$$

The Phase B component is the difference between these two equations; thus, Equation B-4 minus Equation B-3 is:

$$0 = T_k'(Y_{c_k} + Y_{k_m} + Y_{s_k}) - T_k(Y_{c_k} + Y_{k_m} + Y_{s_k}) - T_m'Y_{k_m} + T_mY_{k_m} \quad (\text{B-5})$$

This can be written:

$$0 = \Delta T_k(Y_{c_k} + Y_{k_m} + Y_{s_k}) - \Delta T_m(Y_{k_m}) \quad \dots \quad (\text{B-6})$$

where

$$\Delta T_k, \Delta T_m = (T_k' - T_k), (T_m' - T_m) \text{ etc.}$$

Equation B-6 is the same as is obtained by putting  $T_s$  in Equation B-4 at ground potential. Then  $T_k'$  would represent an increase in temperature above ground potential. In this way, the separate contribution of any load-producing element can be obtained.

## APPENDIX C

### APPROXIMATIONS DUE TO MAKING $T_g$ INDEPENDENT OF THE THERMAL CIRCUIT

Recall that  $T_g$  is connected to  $g$  on the inner surface of the glass and that its variation throughout the cycle was fixed as a result of calculations made in deriving the  $T_g$  values from Tables 14 and 21 of Chapter 13 of THE GUIDE 1954. The heat gain data in these tables were based upon the postulate that both room air and room surface temperatures remain fixed at 75 F. This is true for the room air but not for the surfaces, as Phase A results show. Some adjustments in the Phase B results were therefore made.

In a practical situation then, the glass temperatures would be greater than the values given by the curves of Fig. 5 of the first paper (Reference 1), which were based upon both room air and room surfaces remaining constant at 75 F. This is true because the glass temperature must be such as to be in equilibrium with both the indoor and outdoor environments. Indeed, this is one of the points to be demonstrated by circuit principles and could have been handled by providing a sol-air temperature generator for the glass. Though this could have easily been done for the homogeneous sheet glass, the sol-air temperature concept has not yet been successfully applied to glass block. Hence, the simplification used in Phase A seemed justified and has been carried over to Phase B.

In order to check the adequacy of thus treating the glass circuit branch, window glass temperatures were recomputed. For Phase A, the room surface temperatures as found by solving the equations for the Phase A conditions, together with the constant room air temperature of 75 F, yielded window glass temperatures for Case I ranging from zero to 0.4 deg higher than the values actually used in solving the circuit equations. Case II differences would be less. These discrepancies were sufficiently small that the load current from the glass to point  $a$  and the various room surface temperatures and their resultant load currents were judged to be essentially correct without adjustments.

In Phase B, point  $g$  was put at ground potential, that is, equal to the room air temperature, whereas actually it rises under the influence of the other room surfaces, which were warmed by the sun, and contributes a load current to point  $a$ . This temperature rise was taken into account by writing a heat balance on point  $g$ . The sum of the net radiation currents from the other room surfaces was equated to the thermal currents to  $a$  from point  $g$  and to the outdoors from point  $g$ . These thermal currents are included in the results as a small load increment and a heat flow current to the outdoors. A very small residual error which remained because of the secondary interaction of the glass temperatures with the other surface temperatures was ignored.

## APPENDIX D

TABLE D-1—PROPORTIONING—SOLAR RADIATION CURRENTS

TRANSMITTED DIFFUSE RADIATION, $Q_d$		DIRECT OR BEAMED RADIATION, $Q_D$		
		SUN TIME	FRACTIONAL CROSS SECTION OF DIRECT BEAM	
			ENTERING ROOM	EXCLUDED FROM ROOM
To wall b	0.056	5 a.m.	0.694	0.306
To ceiling c	0.330	6	0.268	0.732
To wall d	0.056	7	0.000	1.000
To wall e	0.264	...	—	—
To floor f	0.294	5 p.m.	0.000	1.000
		6	0.268	0.732
		7	0.694	0.306

TABLE D-2—HARMONIC COEFFICIENTS OF CURVES OF FIG. 3

STEADY STATE HARMONIC		WINDOW GLASS—CASE I			GLASS BLOCK— CASE II
		TRANSMITTED DIFFUSE RADIATION	TRANSMITTED DIRECT RADIATION ON:		TRANSMITTED DIFFUSE RADIATION
			WALL b	WALL d	
			$Q_d$	$Q_D$	
	$a_0$	+505.0	+16.1	+16.1	+173.0
1	$a_1$	-604.0	+2.8	+2.8	-219.0
	$b_1$	0	-31.6	+31.6	0
2	$a_2$	+36.4	-30.4	-30.4	+13.8
	$b_2$	0	-4.6	+4.6	0
3	$a_3$	+121.0	-6.4	-6.4	+45.9
	$b_3$	0	+29.2	-29.2	0
4	$a_4$	+42.8	+27.0	+27.0	+11.8
	$b_4$	0	+8.0	-8.0	0
5	$a_5$	-32.8	+9.0	+9.0	-22.0
	$b_5$	0	-24.6	+24.6	0
6	$a_6$	-30.8	-22.4	-22.4	-10.2
	$b_6$	0	-8.8	+8.8	0
7	$a_7$	+29.02	-9.2	-9.2	+9.0
	$b_7$	0	+19.6	-19.6	0
8	$a_8$	+27.6	+16.8	+16.8	+8.6
	$b_8$	0	+8.8	-8.8	0
9	$a_9$	-9.6	+8.4	+8.4	-3.0
	$b_9$	0	-14.0	+14.0	0
10	$a_{10}$	-20.0	-11.6	-11.6	-6.2
	$b_{10}$	0	-7.2	+7.2	0.0

Fourier-series expressions of the curves of Fig.3 have the following form:

$$Q = a_0 + a_1 \cos 15 \theta + b_1 \sin 15 \theta + a_2 \cos 30 \theta + b_2 \sin 30 \theta + a_3 \cos 45 \theta + b_3 \sin 45 \theta + \dots, \text{ Btu per hour}$$

where

$\theta$  = elapsed time in hours after midnight, suntime 15  $\theta$ , 30  $\theta$ , 45  $\theta$ , etc., are equivalent angular degrees for first, second, third harmonics respectively.

$a_0$  = 24 hr. mean value.

$a_1, a_2, a_3$ , etc. = coefficients of cosine terms for first, second, third, etc., harmonics.

$b_1, b_2, b_3$ , etc. = coefficients of sine terms for first, second, third, etc., harmonics.

Note: The corresponding vector notation for each harmonic =  $a - jb$ .

TABLE D-3—EQUATIONS TO BE SOLVED FOR TEMPERATURES PRODUCED BY DIRECTLY TRANSMITTED RADIATION  
Phase B—Case I—Window Glass

CIRCUIT POINT	$T_b$	$T_d$	$T_h$	$T_o$	$T_e$	$T_f$	$T_m$	$T_k$	$Q_d^a$	$Q_b^b$
$c$	$25.2 T_b$	$+ 25.2 T_d$	$+ 35.5 T_h$	$+ 55.5 T_o$	$+ C$	$T_e$	$0$	$0$	$= -0.330 Q_d$	$+ 0$
$d$	$5.2 T_b$	$+ 17.4 T_d$	$+ 14.0 T_h$	$+ 17.4 T_o$	$+ 25.2 T_e$	$+ 25.2 T_f$	$0$	$0$	$= -0.056 Q_d$	$+ Q_{b,d}$
$e$	$17.4 T_b$	$+ 17.4 T_d$	$+ 24.2 T_h$	$+ 17.4 T_o$	$+ 55.5 T_e$	$+ 55.5 T_f$	$0$	$0$	$= -0.264 Q_d$	$+ 0$
$f$	$25.2 T_b$	$+ 25.2 T_d$	$+ 38.0 T_h$	$+ 55.5 T_o$	$+ 117.6 T_e$	$+ F$	$0$	$0$	$= -0.294 Q_d$	$+ 0$
$h$	$14.0 T_b$	$+ 14.0 T_d$	$+ 97.1 T_h$	$+ 24.2 T_o$	$+ 35.5 T_e$	$+ 38.0 T_f$	$+ 97.1 T_m$	$+ 73.4 T_k$	$= +0$	$+ 0$
$m$	$0$	$0$	$0$	$0$	$0$	$0$	$+ M$	$+ K$	$= +0$	$+ 0$
$k$	$0$	$0$	$0$	$0$	$0$	$0$	$+ 73.4 T_m$	$+ 0$	$= +0$	$+ 0$
$b$	$B$	$+ 5.2 T_d$	$+ 14.0 T_h$	$+ 17.4 T_o$	$+ 25.2 T_e$	$+ 25.2 T_f$	$+ 0$	$0$	$= -0.056 Q_d$	$+ Q_{b,b}$

Values of the complex quantities B, C, D, E, F, H, M, K:

HARMONIC	B	C	D	E	F	H	M	K
$a_0$	-157	-399	-157	-326	-609	-317	-171	-368
1	-160	-446	-160	-330	-656	-317	-171	-368
2	-166	-570	-166	-342	-780	-317	-171	-368
3	-175	-102	-175	-362	-946	-317	-171	-368
4	-188	-133	-188	-389	-1117	-317	-171	-368

<sup>a</sup> See Table D-2 for  $Q_d$  values.<sup>b</sup> See Table D-2 for  $Q_{b,d}$  values.

Note: Admittances B and D are identical.

TABLE D-4—EQUATIONS TO BE SOLVED FOR TEMPERATURES PRODUCED BY DIRECTLY TRANSMITTED RADIATION  
Phase B—Case II—Glass Block

CIRCUIT POINT	$T_d$	$T_h$	$T_e$	$T_c$	$T_f$	$T_m$	$T_k$	$Q_d^a$
$c$	$50.4 T_d$	$+ 35.5 T_h$	$+ 55.5 T_e$	$+ C T_c$	$+ 117.6 T_f$	$+ 0$	$+ 0$	$= -0.330 Q_d$
$d$	$D T_d$	$+ 14.0 T_h$	$+ 17.4 T_e$	$+ 25.2 T_c$	$+ 25.2 T_f$	$+ 0$	$+ 0$	$= -0.112 Q_d$
$e$	$34.8 T_d$	$+ 24.2 T_h$	$+ E T_e$	$+ 55.5 T_c$	$+ 55.5 T_f$	$+ 0$	$+ 0$	$= -0.264 Q_d$
$f$	$50.4 T_d$	$+ 38.0 T_h$	$+ 55.5 T_e$	$+ 117.6 T_c$	$+ F T_f$	$+ 0$	$+ 0$	$= -0.294 Q_d$
$h$	$28.0 T_d$	$+ H T_h$	$+ 24.2 T_e$	$+ 35.5 T_c$	$+ 38.0 T_f$	$+ 97.1 T_m$	$+ 0$	$= +0$
$m$	$0$	$+ 97.1 T_h$	$+ 0$	$+ 0$	$+ 0$	$+ M T_m$	$+ 73.4 T_k$	$= +0$
$k$	$0$	$+ 0$	$+ 0$	$+ 0$	$+ 0$	$+ 73.4 T_m$	$+ K T_k$	$= +0$

Values of the complex quantities C, D, E, F, H, M, K:

HARMONIC	C	D	E	F	H	M	K
$d_0$	-399	-152	-326	-609	-317	-171	-368
1	-446	-155	-330	-656	-317	-171	-368
2	-570	-161	-342	-780	-317	-171	-368
3	-735	-170	-362	-946	-317	-171	-368
4	-906	-183	-389	-1117	-317	-171	-368

<sup>a</sup> See Table D-2 Col. 4 for  $Q_d$  values; note that there are no  $Q_d$  values in Case II.Note:  $T_h$  and  $T_d$  are identical in Case II of Phase B.

TABLE D-5—SOLUTIONS OF THE CIRCUIT EQUATIONS OF TABLES D-3 AND D-4

Harmonic coefficients give temperature increments for points in circuit due to transmitted solar radiation  
 $T = a_0 + a_1 \cos 15\theta + b_1 \sin 15\theta + a_2 \cos 30\theta + b_2 \sin 30\theta + a_3 \cos 45\theta + b_3 \sin 45\theta + a_4 \cos 45\theta + b_4 \sin 45\theta + \dots, \text{ } ^\circ\text{F}$  deg  
 $\theta = \text{elapsed time in hours from midnight}$

where

CASE I	$a_0$	$a_1$	$b_1$	$a_2$	$b_2$	$a_3$	$b_3$	$a_4$	$b_4$
$T_b$	0.6227	-0.3196	-0.4087	-0.1366	-0.0862	-0.0625	+0.1433	+0.0859	+0.1128
$T_d$	0.6227	-0.3973	-0.0438	-0.1553	-0.0401	+0.0763	-0.1021	+0.1246	+0.0566
$T_e$	0.7803	-0.4411	-0.3898	+0.0091	+0.0015	+0.0328	+0.0383	+0.0122	+0.0174
$T_f$	0.7229	-0.6049	-0.2720	+0.0152	-0.0003	+0.0685	+0.0528	+0.0222	+0.0285
$T_i$	0.5302	-0.3408	-0.2461	+0.0064	-0.0003	+0.0283	+0.0280	+0.0101	+0.0138
$T_h$	0.3185	-0.1643	-0.1301	+0.0098	-0.0059	+0.0126	+0.0139	+0.0132	+0.0136
$T_k$	0.0394	Values very small							
$T_m$	0.1979	+0.0119	-0.0530	-0.0002	-0.0001	-0.0013	+0.0014	-0.0008	+0.0011

CASE II	$a_0$	$a_1$	$b_1$	$a_2$	$b_2$	$a_3$	$b_3$	$a_4$	$b_4$
$T_b$	0.1732	-0.1357	-0.0807	+0.0054	+0.0050	+0.0131	+0.0145	+0.0026	+0.0032
$T_d$	0.1732	-0.1354	-0.0807	+0.0054	+0.0050	+0.0131	+0.0145	+0.0026	+0.0032
$T_e$	0.2592	-0.1605	-0.1411	+0.0051	+0.0064	+0.0126	+0.0153	+0.0028	+0.0030
$T_f$	0.2401	-0.2191	-0.0980	+0.0101	+0.0067	+0.0264	+0.0212	+0.0055	+0.0050
$T_i$	0.1761	-0.1229	-0.0904	+0.0041	+0.0045	+0.0109	+0.0113	+0.0024	+0.0023
$T_h$	0.1024	-0.0471	-0.0362	+0.0023	+0.0023	+0.0058	+0.0062	+0.0012	+0.0013
$T_k$	0.0128	Values very small							
$T_m$	0.0638	+0.0032	-0.0151	-0.0002	+0.0004	-0.0005	-0.0004	-0.0001	+0.0001

## DISCUSSION

S. F. GILMAN, (Syracuse, N. Y.): This is another excellent paper from the ASHAE Laboratory and is of great interest to those concerned with load estimating.

I would like to point out that although the procedure may appear quite involved, it can be reduced to routine, simple calculations. Expressing inputs as Fourier series is easily accomplished by filling in a table with a set of given numerical values, and then performing only arithmetical operations. For studies in which the room temperature is held constant, as is the case in this paper, the equations that result have a very orderly pattern. The matrix of the coefficients is symmetrical about the principal diagonal; hence, cross-diagonal values are easily checked for the same magnitude, sign and position in the matrix. Moreover, values along the diagonal are resistances and appear in sets of two, which differ only in sign. Capacitances take positions around the diagonal and are all preceded by a minus sign. It is therefore evident that signs and positions are easily checked for correctness. In fact, after setting up a few problems, the pattern is such that an entire set of equations can be formulated in a very few minutes.

The next step is to solve the resultant system of  $n$  linear equations in  $n$  unknowns. The laboratory used a special computing machine. This is very fast and convenient, but is not absolutely essential. For a thermal circuit of our residential test facility involving the roof and four walls, we obtained a set of 20 equations. These were solved by the so-called "relaxation method" in eight hours with the type of electric calculating machine widely used in business offices. I should mention that the computations were made in our computer section by an experienced operator.

A set of 28 equations were solved by an automatic computer at about the same cost. Economically, we have found little difference between manual and automatic computer solutions. Therefore, anyone interested in thermal circuit analysis should not feel that an electronic computer is absolutely necessary for conducting such work.

I would like the authors' opinions on some of the assumptions. The first is that the transmitted portion of the diffuse radiation emanated uniformly from the glass surface. I fully realize that we don't know the answer exactly, but the opinion of the authors as to how close they feel this assumption may be to reality would be helpful. The same applies to the assumption that the room surfaces completely absorbed the solar radiation incident upon them.

Of great practical importance is the case of a venetian blind with slats set 45 degrees upwards. Starting with a sol-air temperature, would the authors outline the circuit setup and give estimates of the amounts of radiation absorbed and reflected by the slats. Would the blind tend to distribute more radiation on the ceiling and floor, and less on the walls?

The possibility of additional thermal storage effect by allowing the room temperature to rise is pointed out in the paper. This is an important topic for research. A rise in room temperature of as little as three degrees can conceivably provide as much or more thermal storage than that obtained from radiation effects. This condition is handled by letting the room temperature remain as a variable in the equations. From the solution is obtained the temperature rise and the time that it reaches a maximum. However, the case of varying room temperature is considerably more difficult to handle with the thermal circuit technique, especially if interior furnishings are considered. Moreover, when the room temperature is allowed to rise, the flow of heat into the walls, and partitions is governed primarily by the convection component of the surface conductance coefficient—and values for different surfaces in air conditioned spaces can only be crudely estimated. The importance of accelerating research in this direction cannot be overemphasized. In addition, the time has come to begin a field study to relate thermal circuit solutions to actual equipment operation. That such a field study will require precise instrumentation and careful analysis is borne out by Fig. 6, which shows surface temperature variations of less than three degrees. Accurate evaluation of these will be very difficult. Simple structures in which conditions can be accurately controlled and measured should evidently be attacked first.

As a final comment, it is hoped that both the analytical and experimental phases of this project can be accelerated. It is a difficult problem, but the results can provide much valuable information.

**AUTHOR'S CLOSURE (G. V. Parmelee):** A major purpose of the activities in thermal-circuit analyses at the ASHAE Research Laboratory is to bring forth principles and methods for application by others. We are warmly pleased to commend the extensions and applications which have been made by the industry.

Indeed, our major concern at present is how best to expedite application, wherein the development of broad experience, and the sharing of this experience, can lead to savings in time and money for all of the many problems which are amenable to circuit treatment. Analyses of periodic heating and cooling loads are only one type of problem. The broad plans include, in addition to these, the following:

1. Analysis of transient and aperiodic phenomena.
2. Automatic controls, components and systems.
3. Equipment characteristics in time-variable operation.
4. Source characteristics in time-variable operation. This includes the human body and a correlation with comfort.
5. Moisture-transfer systems, all aspects.
6. Evaluation of the money, time, results, reasonable-simplification, and accuracy aspects of circuit or system analyses.
7. Extension to non-linear systems.
8. Analysis of operating schedules and economics.
9. Ultimately, when and if justified, an analysis center for circuit problems and the producing of *practical* calculation tables and charts,—to be operated by the ASHAE Research Laboratory for the benefit of the entire profession.

To complete the picture, the development of thermal circuit techniques was proposed in the ASHAE program back in 1946. The long initial lag in starting this activity was primarily the result of inadequate supporting interest.

This background material has been introduced as a broad reply to that portion of Mr. Gilman's comments which deals with the general program, rather than with the present particular paper.

**1. Numerical Calculations:** We agree with the inherent simplicity of numerical methods. Part I of our work pointed this out.

The computing machine which we used was a standard model for solving simultaneous equations.

**2. Assumptions:** (a) Uniform emission of transmitted diffuse radiation. This was adopted as a simple first approximation, which we felt to be compatible with the demonstrative purpose of our work and with the relatively gross assumption covering the radiation input function. If anyone has reliable better data at hand, the assumption can be improved. (b) Complete absorption by room surfaces. We have set up circuits involving inter-reflections, and these really are not too difficult. For present purposes, however, nothing was to be gained from such a refinement.

**3. Venetian Blinds:** This topic is beyond the scope of the present paper,—as are all shading devices. The venetian blind problem has been analyzed, however, in other work without undue difficulty.

**4. Variable Room Air Temperature:** Treatment of variable room air temperature is merely another step in the general method which we have set forth. This problem also has been studied, but it is beyond the scope of the present paper.

**5. Needed Additional Research:** The planned general program includes the topics mentioned,—and more too. Accomplishment is largely dependent upon the nature and magnitude of the supporting interest.

A considerable amount of the work presently being done in the ASHAE Environment Laboratory will provide data on resistances and capacitances for the room portion of a thermal circuit.

The UCLA cooperative project will show a comparison of circuit solutions with test results.



**1530**

## GAS IS AN IMPORTANT FACTOR IN THE THERMAL CONDUCTIVITY OF MOST INSULATING MATERIALS†

### Part II\*

By R. M. LANDER\*\*, MINNEAPOLIS, MINN.

THE THERMAL conductivity of insulating materials is dependent upon four modes of heat transfer: gas conduction, convection, radiation, and solid conduction. Although the general principles relating to each of these modes of heat transfer are fairly well understood, their combined effect on heat transfer through insulating materials is very complicated. Since insulating materials are composed largely of gas, the mechanism of heat transfer through the gas has an important influence on the overall conductivity coefficient.

The thermal conductivity of gases increases with temperature and in general decreases as the molecular weight is increased. The conductivity of gases is independent of pressure from one atmosphere down to moderately low pressures. Convection in insulating materials is a complex process and depends upon the prevailing temperature, pressure and kind of ambient□ gas, and the density, dimensions, and orientation of the insulation specimen. The experimental method used in this investigation does not directly differentiate between gas conduction and convection, and the term *gas transferred heat* will be used to designate the total heat transferred by the gas within the insulating material.

Heat transferred by radiation through the gas spaces of insulating materials is

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† This paper covers a part of the results of a research program conducted by the Engineering Experiment Station and supported in part by the Graduate School, University of Minnesota.

\* Part I of this paper appeared in ASHVE TRANSACTIONS, Vol. 58, 1952, p. 155.

\*\* Research Fellow University of Minnesota.

□ Ambient gas is the gas occupying the volume not filled by the solid portion of the insulating material. Presented at the 61st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1955.

independent of the ambient gas. When the percent volume occupied by the solid is small, heat transferred by radiation is approximately inversely proportional to insulation density and varies directly as the cube of the mean temperature. In some cases, the transparency of solid material to radiation is also a function of mean temperature.



FIG. 1. APPARATUS ASSEMBLED IN POSITION FOR TEST  
SHOWING EXTERIOR OF STEEL VACUUM CHAMBER A

The introduction of a solid material into a gas-filled space ordinarily reduces heat transfer by means of radiation and convection but increases gas transferred heat. This increase in gas transferred heat arises from the displacement of the gas with material of a higher conductivity and by an increase in the heat transfer area between the solid and ambient gas.

The heat transferred through the solid portion of the insulation depends upon the kind of material and the arrangement and size of the fibers or particles. Division of a solid material into a finely divided state can reduce the conductivity to a

value lower than that of still air, by reducing the heat path<sup>‡</sup> below the mean free path of the gas molecules. For large volume ratios of gas to solid material, solid conduction is approximately proportional to density.

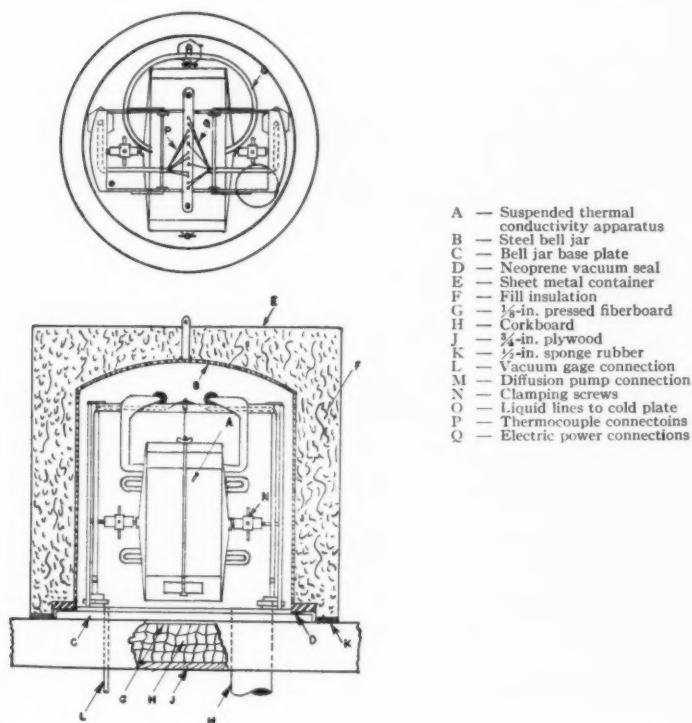


FIG. 2. DIAGRAMMATIC SKETCH OF  
VACUUM CHAMBER AND GUARDED  
HOT PLATE APPARATUS

Recent papers<sup>1,2</sup> have shown that the major mode of heat transfer through most insulating materials is by gas transferred heat. Laboratory determination of this quantity is arrived at by difference, subtracting the conductivity obtained with gas extracted from that obtained with gas present. This research paper

<sup>‡</sup> Heat path is defined as the mean distance across confining spaces in an insulating material.

<sup>1</sup> Exponent numerals refer to References.

describes the investigation of the reduction in thermal conductivity of some common insulating materials by replacing the ambient air with gases of larger molecular weight. Experimental data will be presented for several materials showing the relative importance of heat transferred by radiation and by conduction through the solid material.

#### DESCRIPTION OF APPARATUS

The apparatus used in this investigation is shown in Figs. 1 and 2. Fig. 1 shows the assembled apparatus with the insulated bell jar lowered over the conductivity apparatus in position for test. The mechanical fore pump and oil diffusion pump for evacuating the vacuum chamber to an ultimate vacuum of 0.1 micron of mercury are shown underneath the table. At the left next to the wall is a vacuum gage and the constant-temperature alcohol bath, which supplies the cold plates with alcohol at any temperature between 100 F and -75 F. A five-ton monochlorodifluoromethane (F-22) condensing unit, not shown, provides the necessary refrigerating capacity.

Fig. 2 is a diagrammatic sketch of the pertinent details of the apparatus. A neoprene gasket is used to seal the bell jar to the base plate. All thermocouple leads, electric power leads, liquid lines for the cold plates, gage connections and vacuum pump connections are brought through vacuum-tight seals in the base plate.

The 12-in. suspended guarded hot plate contained within the bell jar was built on the same principle as previously developed at this laboratory with some refinements to adapt it to the special program. The actual test area is 8 in. square with a 2-in. guard ring.

Auxiliary equipment consists of dc power supply, switchboard, precision potentiometer, and sensitive temperature controllers.

#### PROCEDURE

In making thermal conductivity determinations on materials at atmospheric pressure with gases other than air, the pressure in the vacuum chamber is reduced to the limiting pressure of the fore pump (about 5 microns of mercury) and the chamber is then filled with the gas under study. Mixtures of gases are obtained by filling the vacuum chamber to the partial pressure required to give the desired proportions by volume.

All materials were tested at the highest obtainable vacuum (about 0.1 micron of mercury absolute pressure) in order to reduce gas transferred heat to a minimum. All tests were conducted with a 50 deg temperature difference across the test specimen and at a mean temperature of 100 F, except as noted.

#### MATERIALS TESTED

Because of the large amount of time required to make precise thermal conductivity measurements, initial tests were limited to a relatively few materials whose characteristics were representative of the major classes of insulating materials. The materials selected for test were corkboard, cotton, glass wool, hair felt, kapok, mineral wool, silica aerogel, wood fiber, and wood fiberboard. In addition to these, a special test specimen was constructed by using eight sheets of aluminum foil between nine sheets of corrugated paper, the corrugations of alternate layers

TABLE 1—REDUCTION IN HEAT TRANSFER COEFFICIENTS OF VARIOUS MATERIALS BY REPLACING THE AIR WITH GASES OF HIGHER MOLECULAR WEIGHT  
(100 F Mean Temperature and 50 Deg Temperature Difference)

MATERIAL	DENSITY, LB/CU FT	THICK- NESS, IN.	THERMAL CONDUCTIVITY, <i>k</i>				AIR TRANS- FERRED HEAT, BTU (IN.)/ (HR) (SQ FT) (F DEG)	PERCENT REDUCTION			
			F-12		LOW PRESS.	TOTAL HEAT		GAS TRANS HEAT			
			AIR	CO <sub>2</sub>		CO <sub>2</sub>			F-12		
Corkboard.....	6.67	1	0.283	0.222	0.195	0.106	0.177	21.6	31.1	34.5	49.7
Corkboard.....	12.4	1	0.314	0.256	0.218	0.120	0.194	18.5	30.6	29.9	49.5
Cotton.....	0.78	1	0.288		0.164	0.072	0.216		43.1		57.4
Cotton.....	1.53	1	0.257	0.192	0.1365	0.052	0.205	25.3	46.9	31.7	58.8
Cotton.....	2.62	1	0.248	0.180	0.120	0.033	0.215	27.4	51.0	31.6	59.5
Coarse Glass Wool.....	2.8	1	0.267	0.197	0.141	0.055	0.212	26.2	47.2	33.0	59.4
Fine Glass Wool "A".....	0.58	1	0.2505	0.183	0.133	0.050	0.2005	26.9	46.9	33.7	58.6
Fine Glass Wool "A".....	1.94	1	0.2145	0.148	0.097	0.0195	0.195	31.0	54.8	34.1	60.2
Fine Glass Wool "A".....	3.80	1	0.211	0.146	0.093	0.012	0.199	30.8	55.9	32.7	59.3
Fine Glass Wool "B".....	0.78	1	0.302	0.230	0.1715	0.0665	0.2355	23.8	43.2	30.6	55.4
Hair Felt.....	11.1	3/4	0.260	0.185	0.124	0.031	0.229	28.8	52.3	32.8	59.4
Kapak.....	0.26	1	0.366	0.294	0.233	0.091	0.275	19.7	36.3	26.2	48.4
Kapak.....	0.99	1	0.2445	0.1765	0.1235	0.0375	0.207	27.8	49.5	32.9	58.0
Kapak.....	3.96	1	0.244	0.169	0.115	0.0175	0.2265	30.7	52.9	33.1	57.5
Mineral Wool.....	7.7	1	0.258	0.184	0.126	0.037	0.221	28.7	51.2	33.5	59.7
Silica Aerogel.....	8.6	1	0.1715	0.1235	0.0255	0.0255	0.146	28.0	28.0		32.9
Special Specimen.....	3.1	1	0.231	0.1635	0.1125	0.0145	0.2165	29.2	51.3	31.2	54.7
Wood Fiberglass.....	15.5	3/4	0.353	0.273	0.210	0.075	0.278	22.7	40.5	28.8	51.4
Wood Fiber.....	3.5	1	0.2715	0.1425	0.052	0.052	0.2195	47.5	47.5		58.8
Wood Fiber.....	6.9	1	0.277	0.202	0.142	0.033	0.244	27.1	48.7	30.7	55.3

being at right angles to each other. Air spaces varying in width from  $\frac{1}{4}$  in. to 1 in. and bounded by the copper plates of the test apparatus were also tested.

Corkboard and wood fiberboard were oven dried before testing. In the test they were clamped between the plates and the thickness of the specimen was measured as set up for test. All loose and flexible materials were tested on an *as received* basis, the proper thicknesses being maintained by wood frames around the specimen edges or by special space blocks placed in the corners of the test

TABLE 2—EFFECT OF REDUCING THE GAS PRESSURE ON THE THERMAL CONDUCTANCE OF COPPER BOUNDED SPACES FILLED WITH VARIOUS GASES  
(100 F Mean Temperature and 50 Deg Temperature Difference)

0.990-IN. GAS SPACE					
AIR		CARBON DIOXIDE		F-12	
P	a	P	a	P	a
29.3	0.581	28.7	0.552	28.9	0.604
19.8	0.490	18.9	0.472	20.0	0.514
10.0	0.375	8.7	0.355	9.8	0.399
0.5	0.333	0.5	0.277	1.0	0.247

0.251-IN. GAS SPACE					
AIR		CARBON DIOXIDE		F-12	
P	a	P	a	P	a
29.3	0.882	29.5	0.652	28.8	0.643
19.3	0.883	18.7	0.635	18.5	0.500
8.9	0.882	8.6	0.634	8.8	0.458
0.5	0.891	0.55	0.634	0.5	0.445

P = Pressure, inches mercury.

a = Conductance, Btu per (square foot) (hour) (Fahrenheit degree).

specimen. In order to get a good test sample, the kapok and loose wood fiber materials were repacked. The other samples were tested without repacking.

### TEST RESULTS

Table 1 lists thermal conductivity values obtained for the insulating materials selected for test, using three different ambient gases. The reduction in thermal conductivity using carbon dioxide instead of air ranges from 18 to 31 percent and by using dichlorodifluoromethane (F-12) from 28 to 56 percent.

Consideration of the gas transferred heat alone, as removed from that transferred by radiation and solid conduction, gives a better understanding of the ambient gas effect. The percent reduction in gas transferred heat using carbon dioxide instead of air is from 26 to 34 percent and for F-12 is from 33 to 60 percent.

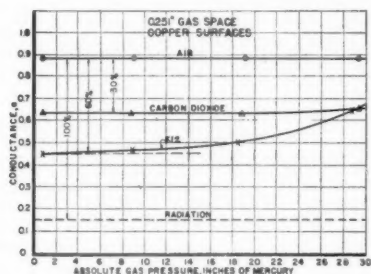


FIG. 3. THERMAL CONDUCTANCE OF  $\frac{1}{4}$ -IN. GAS SPACE AS A FUNCTION OF VARIOUS AMBIENT GASES AND PARTIAL ATMOSPHERIC PRESSURES

Air transferred heat for the three fibrous materials, cotton, glass wool "A", and kapok, is greatest for the low insulation densities, declines to a minimum for the medium insulation densities and then increases again as density is raised to a high level. The high rate of air transferred heat for the low densities may be attributed to convection. Since the conductivity of all three materials with the air removed continued to decrease with increasing density, the increasing rate of air transferred heat for the high density materials must be due to *bypassing* of heat by increasing the transfer area between solid and ambient gas.

Low density corkboard exhibited next to the lowest rate of air transferred heat but had the highest rate of solid conduction. This explains the reported low conductivity value of expanded corkboard. The very low rate of gas transferred heat shown by silica aerogel, the single material having a rate less than that of low density corkboard, may be attributed to its finely divided state.

Thermal conductivity values for  $\frac{1}{4}$  in. and 1 in. wide gas spaces filled with air, carbon dioxide, and F-12 at various pressures are listed in Table 2, and illus-

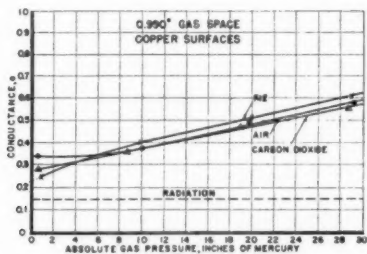


FIG. 4. THERMAL CONDUCTANCE OF 1-IN. GAS SPACE AS A FUNCTION OF VARIOUS AMBIENT GASES AND PARTIAL ATMOSPHERIC PRESSURES

trated in Figs. 3 and 4. Kinetic theory predicts that the conductance of these gas spaces should be independent of gas pressure, providing heat transfer by convection is negligible. Deviation from this prediction for the 1-in. gas space shows

TABLE 3—REDUCTION IN THERMAL CONDUCTIVITY OF SEVERAL MATERIALS BY PARTIALLY REPLACING THE AIR WITH GASES OF HIGHER MOLECULAR WEIGHT  
(100 F Mean Temperature)

PERCENT OF GAS IN AIR	THERMAL CONDUCTIVITY, $k$			
	KAPOK, 0.26 LB/CU FT		GLASS WOOL "A" IN F-12	
	CARBON DIOXIDE	F-12	0.58 LB/CU FT	3.8 LB/CU FT
0	0.361	0.361	0.250	0.211
25	0.346	0.307	0.202	0.162
50	0.324	0.270	0.170	0.131
75	0.309	0.247	0.148	0.109
100	0.292	0.228	0.133	0.093

that convection is an important mode of heat transfer for all three gases over the entire pressure range. The results for the  $\frac{1}{4}$ -in. gas space show that convection is negligible throughout the range of test pressures when the space is filled with air. With carbon dioxide, convection subsides at a very moderate reduction in pressure, while with F-12, a large reduction in pressure occurs before convection subsides.

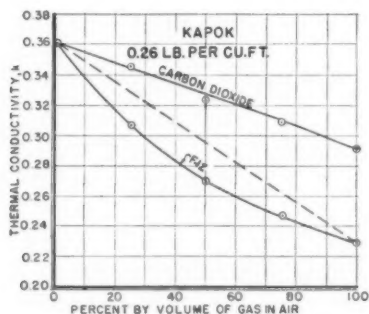


FIG. 5. EFFECT OF VARIOUS PROPORTIONS OF CARBON DIOXIDE AND F-12 IN AIR ON THE THERMAL CONDUCTIVITY OF LOW DENSITY KAPOK

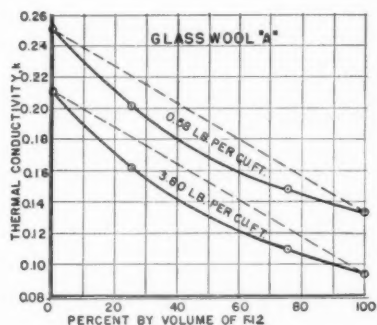


FIG. 6. EFFECT OF VARIOUS PROPORTIONS OF F-12 IN AIR ON THE THERMAL CONDUCTIVITY OF TWO DENSITIES OF GLASS WOOL

The conductances of the  $\frac{1}{4}$ -in. gas spaces as derived from Fig. 3 for a condition of no heat transfer by free convection are 0.882, 0.634 and 0.445 for air, carbon dioxide, and F-12 respectively. The gas conducted heat quantities for the three gases are 0.732, 0.484, and 0.295 respectively as determined by subtracting the heat transferred by radiation (0.150) from the previous values. Thus, the reduction in gas transferred heat obtained by substitution of carbon dioxide for air is

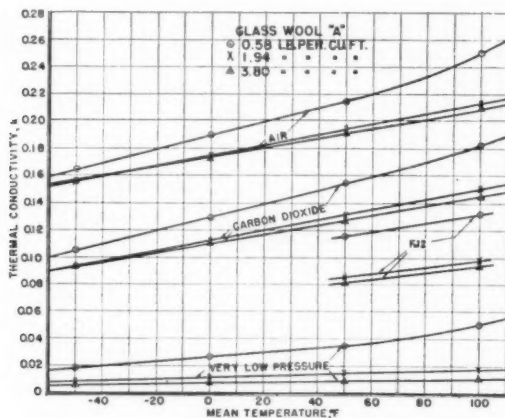


FIG. 7. THERMAL CONDUCTIVITY OF GLASS WOOL AS A FUNCTION OF MEAN TEMPERATURE, AMBIENT GAS, AND DENSITY

TABLE 4—EFFECT OF MEAN TEMPERATURE ON THE THERMAL CONDUCTIVITY OF GLASS WOOL AND FIBERBOARD WITH VARIOUS AMBIENT GASES

MATERIAL	MEAN TEMP., F	THERMAL CONDUCTIVITY, k				GAS TRANSFERRED HEAT (G.T.H.)				
		AIR	CARBON DIOXIDE	F-12	LOW PRESS.	AIR G.T.H.	CO <sub>2</sub> G.T.H.	CO <sub>2</sub> % DECREASE	F-12 G.T.H.	F-12, % DECREASE
1-in. Fine Glass Wool "A", 0.58 lb per cu ft	100	0.2505	0.183	0.133	0.049	0.2015	0.134	33.5	0.084	58.3
	50	0.2155	0.154	0.107	0.0355	0.180	0.1185	34.2	0.0715	60.3
	0	0.190	0.1285		0.0255	0.1645	0.103	37.4		
	-50	0.1645	0.107		0.018	0.1465	0.089	39.2		
1-in. Fine Glass Wool "A", 1.94 lb per cu ft	100	0.214	0.150	0.0985	0.019	0.195	0.131	32.8	0.0795	59.2
	50	0.194	0.130	0.085	0.014	0.180	0.116	35.6	0.071	60.6
	0	0.174	0.1125		0.0105	0.1635	0.102	37.6		
	-50	0.156	0.095		0.0075	0.1485	0.0875	41.1		
1-in. Fine Glass Wool "A", 3.8 lb per cu ft	100	0.211	0.1455	0.095	0.012	0.199	0.1335	32.9	0.083	58.3
	50	0.192	0.1275	0.082	0.008	0.184	0.1195	35.1	0.074	59.8
	0	0.175	0.1105		0.0065	0.1685	0.104	38.3		
	-50	0.157	0.0955		0.0045	0.1525	0.091	40.3		
1-in. Fine Glass Wool "A", Average of all densities corrected for fiber vol- ume	100					0.1955	0.131	33.0	0.081	58.6
	50					0.179	0.1165	34.9	0.0715	60.1
	0					0.163	0.1015	37.7		
	-50					0.147	0.088	40.1		
¾-in. wood fiberboard, 15.5 lb per cu ft	100	0.354	0.279	0.212	0.073	0.281	0.206	26.7	0.139	50.5
	50	0.334	0.259	0.196	0.068	0.266	0.191	28.2	0.128	51.9
	0	0.312	0.237		0.062	0.250	0.175	30.0		
	-45	0.293	0.218		0.058	0.235	0.160	31.9		

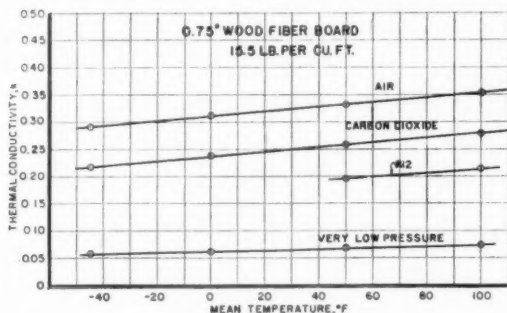


FIG. 8. THERMAL CONDUCTIVITY OF WOOD FIBER-BOARD AS A FUNCTION OF MEAN TEMPERATURE AND AMBIENT GAS

34 percent, and by substitution of F-12 for air is 60 percent. These percentages are identical to the maximum reduction in gas transferred heat for the insulating materials as given in Table 1, and indicate that heat transfer by convection in these materials is a small factor, except for low density kapok.

Results of tests to determine the effect of various mixtures of gas in air on the thermal conductivity of kapok and fine glass wool are listed in Table 3, and illustrated in Figs. 5 and 6. The conductivity of kapok varies linearly with the percent of carbon dioxide present in the air mixture. Results obtained for mixtures of F-12 and air show that F-12 is the predominating gas and gives a greater percentage reduction in conductivity than represented by the percent of F-12 present.

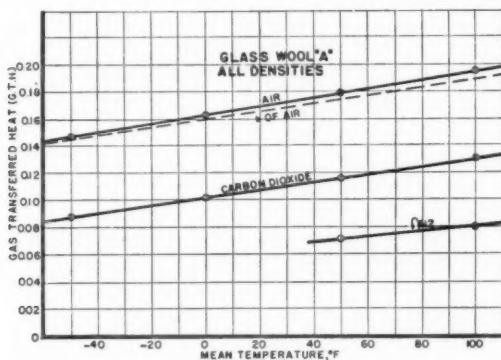


FIG. 9. GAS TRANSFERRED HEAT OF GLASS WOOL FOR VARIOUS AMBIENT GASES AND MEAN TEMPERATURES

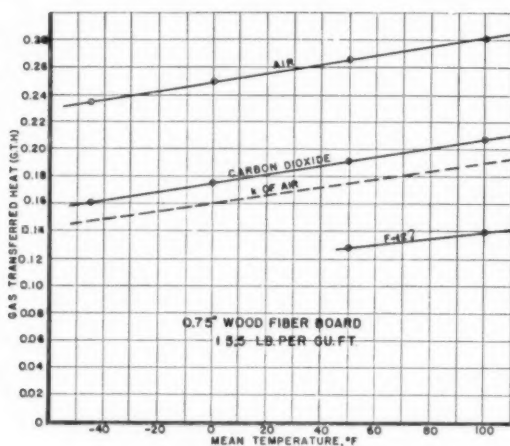


FIG. 10. GAS TRANSFERRED HEAT OF WOOD FIBER-BOARD FOR VARIOUS AMBIENT GASES AND MEAN TEMPERATURES

Results of tests to determine the effect of mean temperature on the conductivity of glass wool "A" and fiberboard are listed in Table 4, and illustrated in Figs. 7 and 8. Table 4 shows that the percent of gas transferred heat increases as the mean temperature is reduced.

TABLE 5—CALCULATED MODES OF HEAT TRANSFER FOR THREE DENSITIES OF GLASS WOOL "A" WITH VARIOUS AMBIENT GASES

AMBIENT GAS	MEAN TEMP.	PERCENT HEAT TRANSFERRED								
		0.58 LB/CU FT			1.94 LB/CU FT			3.80 LB/CU FT		
		G.T.H. <sup>a</sup>	RAD	FIBER	G.T.H. <sup>a</sup>	RAD	FIBER	G.T.H. <sup>a</sup>	RAD	FIBER
Air	100	80.4	19.3	0.3	91.1	8.2	0.7	94.3	4.1	1.6
	50	83.5	16.3	0.2	92.8	6.4	0.8	95.8	2.9	1.3
	0	86.6	13.2	0.2	94.0	5.1	0.9	96.3	2.3	1.4
	-50	89.1	10.7	0.2	95.2	4.2	0.6	97.1	1.6	1.3
Carbon Dioxide	100	73.2	26.4	0.4	87.3	11.7	1.0	91.8	5.8	2.4
	50	76.9	22.8	0.3	89.2	9.6	1.2	93.7	4.3	2.0
	0	80.2	19.5	0.3	90.7	8.0	1.3	94.1	3.6	2.3
	-50	83.2	16.5	0.3	92.1	6.8	1.1	95.3	2.1	2.6
F-12	100	63.2	36.3	0.5	80.7	17.8	1.5	87.4	9.0	3.6
	50	66.8	32.8	0.4	83.5	14.7	1.8	90.3	6.7	3.0

<sup>a</sup> Gas Transferred Heat.

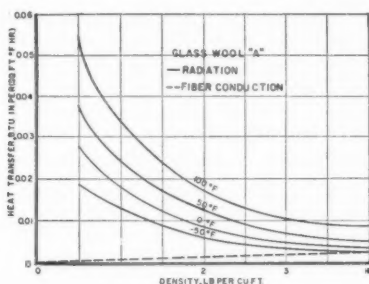


FIG. 11. EFFECT OF DENSITY ON THE HEAT TRANSFERRED BY RADIATION AND FIBER CONDUCTION THROUGH GLASS WOOL AT VARIOUS MEAN TEMPERATURES

Figs. 9 and 10 show that the calculated gas transferred heat for glass wool and fiberboard is a linear function of mean temperature. In making these calculations for glass wool, the gas transferred heat was corrected for the volume occupied by the fibers, which amounts to about 2.5 percent for the highest density glass wool. The generally accepted value for the conductivity of still air is plotted for reference and is about 3 percent lower than the corresponding value of glass wool in air, thus indicating that convection is negligible.

The curves of Fig. 7 show that the conductivity of glass wool is decreased by increasing its density and that the amount of decrease is nearly independent of the ambient gas. The lower group of curves, developed under a test condition of very low pressure, gives the resulting heat transfer due to radiation and fiber conduction, since gas transferred heat is practically eliminated at these low pressures.

In Figs. 11 and 12, radiation has been separated from fiber conduction for the fibrous materials, cotton, glass wool "A", kapok, and wood wool, by considering

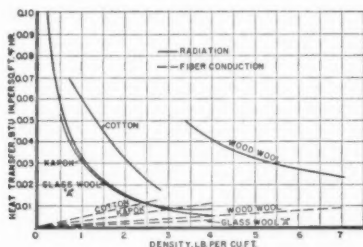


FIG. 12. EFFECT OF DENSITY ON THE HEAT TRANSFERRED BY RADIATION AND FIBER CONDUCTION THROUGH VARIOUS FIBROUS MATERIALS

radiation approximately inversely proportional to density and fiber conduction directly proportional to density. These figures show that for practical insulation densities, radiation is an important mode of heat transfer for all four materials, whereas heat transfer by fiber conduction is negligible.

Calculated values of heat transfer for glass wool with various ambient gases are given in Table 5. Fiber conduction is negligible for all test conditions. Heat transfer by radiation can be an important percentage of the total heat transfer, depending upon insulation density, ambient gas, and mean temperature. Decreasing the mean temperature increases the percent of gas transferred heat because radiation varies as the cube of the mean temperature, whereas gas transferred heat varies linearly with mean temperature.

Since the relationship between conductivity and mean temperature for fiberboard is linear, heat transferred by radiation for this type of material is small. A comparison of the air transferred heat values for fiberboard and glass wool listed in Table 4 shows that air transferred heat is much greater for the fiberboard. This again is attributed to the increased heat transfer area between the solid and ambient gas.

Since gas transferred heat is the major mode of heat transfer in insulating materials at moderate mean temperatures, the reduction of this factor presents the greatest possibility for reducing the thermal conductivity of materials. For special applications, gas transferred heat may be reduced by using gases of low conductivity or by evacuating the gas from the material.

The principle of reducing gas transferred heat by making the length of heat path of the same order of magnitude or smaller than the mean free path of the gas molecules has been demonstrated by a number of investigators. A typical powder utilizing this principle and having a conductivity lower than that of still air at 100 F mean temperature and atmospheric pressure is silica aerogel. In order for present day fibrous materials to utilize this principle, the voids must be greatly reduced in size by decreasing the fiber diameter and perhaps by increasing the insulation density.

For all but one of the materials tested, convection contributed very little to the total heat transferred. This indication of the relatively insignificant effect of convection cannot be accepted for direct application to the practical case, since the isothermal surface temperature of the test specimen and its small size tended to minimize heat transfer by convection. However, it is safe to assume that the effect of convection will not reach important proportions except in the case of large volumes of low density materials.

Considerable heat may be transferred by radiation in low density fibrous materials at 100 F mean temperature. The percent heat transferred by radiation increases rapidly with mean temperature and conversely may become negligible at low mean temperatures. Radiation may be reduced by increasing the insulation density, opacifying the fibers, or coating the fibers with a reflective material.

For common fibrous insulation densities, conduction of heat along the fibers or by fiber contact contributed less than 2 percent to the total heat transferred through the materials tested. The practicability of increasing the insulation density for the purpose of reducing radiation or length of heat path is limited by the increase in fiber conduction and gas transferred heat.

Originally it was intended that this investigation into the modes of heat transfer be continued to cover a greater number of insulating materials over a much larger range of mean temperatures and with greater concentration placed on heat transfer

by means of convection and radiation. This program has been interrupted by the total loss of equipment in the destruction by fire of the Oak Street Laboratory of the University of Minnesota in early 1953.

#### ACKNOWLEDGMENT

The author wishes to express his appreciation to Professor Emeritus F. B. Rowley for many helpful suggestions made during the course of this investigation, and to the Graduate School of the University of Minnesota for its financial assistance.

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3. Thermal Conductivity of Materials by Means of the Guarded Hot Plate (*ASTM Designation C 177-4S*).

#### DISCUSSION

R. S. DILL, Washington, D. C., (WRITTEN): The phenomenon, the main theme of this paper, is not surprising. That gases of higher molecular weight transfer less heat across spaces is well known. The author is to be commended for forcibly bringing the phenomenon to the attention of the profession and for furnishing quantitative data on the effect of substituting heavier gases for air in the interstices of common insulating materials. This has practical possibilities although some obstacles must be overcome to effect its application.

As shown in the tables the thickness of insulation required to attain a specified conductivity can be reduced by roughly one-half if, say, F-12 is substituted for air in some insulations. In addition, if the gas is dry the danger of internal-wall condensation is avoided. In effect the gas is used to expel the air with its accompanying water vapor from the wall. Obviously there are two ways to do this. One is to connect a source of F-12 to the space containing the insulation and to regulate the flow of gas to this space just fast enough to overcome any leaks. The other way is to hermetically seal the walls containing the insulation to retain the gas with which the space is initially charged. Then either the walls must be strong enough to withstand pressure variations or some expansion means such as a bellows, a diaphragm or a balloon must be provided to permit volume changes due to variations in barometric pressure or temperature.

I surmise that these suggestions have occurred to the author and his consultants also and I would like to hear their conclusions concerning the possibilities.

Obviously in the last analysis the utility of the idea must rest on the economics of manufacture or construction of heated or refrigerated enclosures. However, the idea is interesting both from the technical and practical standpoints and merits further exploration in my opinion.

C. F. KAYAN, New York, N. Y., (WRITTEN): This paper represents a nice contribution to the further understanding of the mechanisms of heat transfer in insulation. As

a matter of interpretation, certain questions arise, for which the author could contribute answers:

1. Does the author mean *increase in gas transferred heat* when he refers to the introduction of a solid material into a gas-filled space? This is equivalent to increasing the density of the material, and as a result, less heat will be transferred by the gas than by solid conduction, since in the limit in a solid material no heat will be transferred by gas conduction.
2. What would be the effect of using a smaller temperature difference on the conductance values of F-12 varying with pressure? Since convection seems to play a large role in the change of the conductance of F-12 with pressure, perhaps a different temperature difference would change this value.

W. A. DANIELSON, Raleigh, Tenn., (WRITTEN): The paper makes no mention of surface conduction of the individual spaces within the insulating material. The heat must be transferred from solid material to the gas, across the gas space, then from gas to the solid. With fibrous material the space shape is different from that of cork. While the gas spaces are smaller and consequently with surfaces less than ordinarily dealt with in calculating  $U$ , it does not seem reasonable that this factor can be neglected in research as accurately made and as carefully analyzed as is done in this paper. The authors no doubt can clarify this point.

It would seem logical that there should be greater conductance in gas with heavier molecules as the space filled by these would have a greater density. In solids more dense materials usually have a greater conductance. The authors' theoretical explanation of this condition would be interesting.

B. M. PALMER, \* Newark, Ohio: I, too would like to congratulate the author for adding more of what I consider very good and practical information to the record.

I was quite interested in this paper because for a period of several years we, too, have run data primarily on fibrous insulation of a similar nature and I would say that we have good agreement, and it always makes you feel good when somebody agrees with your data.

I would say that the dilution data between the F-12 and air shows very good agreement.

I would suggest three points for consideration by the author. One: in presenting the reduction in  $k$ , when we replace air with F-12 or  $\text{CO}_2$ , it was shown as a percentage figure. I think it is quite interesting and will give a good rule of thumb if instead of doing that one would subtract and determine what the exact difference is. It will be found that on the one material shown, (take the sample A) from air to F-12 gave an exact amount of drop in each case. So if one knows what an insulation does in air and subtracts a rule-of-thumb figure which is representative of the difference in still air or still gas conduction, the book value in each case, the result is a good rule-of-thumb figure for a typical fibrous insulation.

If this is done one thing will be noticed, though, that in the coarser fibred material where convection is greatest at low densities, this data says that replacing with  $\text{CO}_2$  or F-12 decreases that convection.

The second point—(this is not in criticism, it is just a comment)—at higher density we attribute an increase in conduction due to the increased area between solid and ambient gas and I say that appears applicable.

However, in our experience with fibrous material we have yet to get a crossover at higher densities, and by that I mean that even at higher densities where fibres present a tremendous amount of area, our data has always shown a better  $k$  with a lower fibre diameter. Practically, that is what it means.

\* Owens-Corning Fiberglass Corp.

In the third point, the approach of subtracting vacuum plate data from the standard air hot-plate data to obtain a gas transfer heat value is very good in understanding the problem, but my background for the past several years has been in research and if I switch over to that point of view for a minute I wonder if we dare assume that radiation and fibre conduction are exactly the same whether we have a gas in there or whether we don't.

I am wondering if the curve of temperature through an insulation isn't changed when the conduction phase is present and if that is true could not the radiation factor be modified some there?

I think most people are apt to do what I have done in reading the paper, and that is take our vacuum data, look up in the tables what still air conductivity is, then subtract, and what is left over we say is convection or this bypassing factor; and I say the fallacy there is that we make it rough on our data.

We have accumulated all our errors into one place or if any of our assumptions are incorrect.

Again I say I agree emphatically with all of your conclusions. These are just minor points for consideration.

**AUTHOR'S CLOSURE:** I can add little to the observations of Mr. Dill regarding the difficulty of hermetically sealing an insulating material in an enclosure such as the walls of a domestic refrigerator, but I have no doubt that this difficult problem will be surmounted. One solution to the problem of the distortion of flimsy side walls, due to pressure changes, is to charge the system with a heavy gas such that the walls will always be subject to a slight external pressure, the walls being prevented from collapsing by the compressive strength of the insulation. This method would also be applicable to a completely evacuated system, but the problem of maintaining a good vacuum in a sealed container is much more difficult than sealing in a gas at reduced pressure. A continually evacuated system might be economically feasible for an application in which space is at a premium while the heavy gas idea might be more applicable to small portable enclosures. These are just a few of the many possibilities.

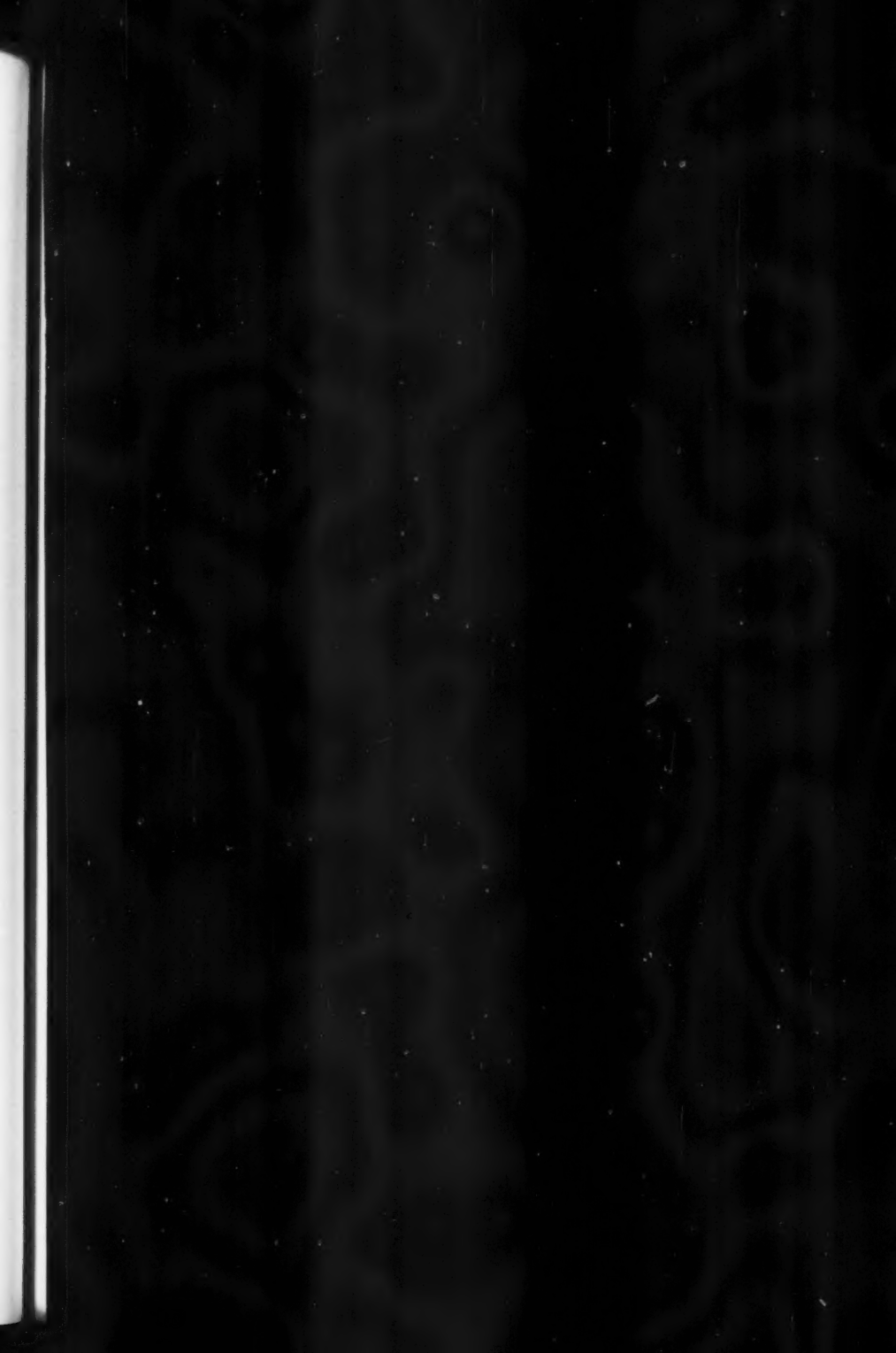
As regards the first question of Professor Kayan, I was thinking in terms of adding the solid material to the gas-filled space in planes normal to the heat path, which is the most effective manner of reducing the heat transfer. This would decrease the length of heat path through the gas, thereby increasing the gas transferred heat. If, on the other hand, the solid material is added as elements parallel to the heat path, the gas transferred heat will be decreased in proportion to the reduced volume of the gas. I have no doubt that the conductance of the gas spaces filled with F-12 at varying pressures would, in most cases, be a function of temperature difference. This might also be the case for some of the materials such as low density kapok. Some of this research was in progress at the time the equipment was destroyed.

Referring to the discussion of Mr. Danielson, the three modes of heat transfer apply equally well to the surface conductance of small air spaces as well as the more familiar surface coefficients for walls. For walls, convection accounts for a considerable portion of the coefficient, whereas this factor is greatly reduced for small air spaces. Increasing the density of such solids as wood or concrete increases the thermal conductivity because the volume of air voids is reduced. Gases of higher molecular weight have a lower conductivity because the molecules move at a much lower velocity than the molecules of gases having a low molecular weight.

I am also gratified that the research work of the laboratories of Mr. Palmer and the University of Minnesota are in such good agreement. The first suggestion of Mr. Palmer and the deduction of the reduction in convection by replacing air in low density materials with heavy gases is very good. It should be kept in mind however, that the difference in thermal conductivity values increases as the fiber volume is increased. In regard to Mr. Palmer's third point, evacuation of the insulation would alter the temperature of the individual fibers somewhat because the primary mode of air trans-

ferred heat would be eliminated. It seems to me that the average temperature gradient through the insulation would remain about the same for the two conditions.

In conclusion I would like to say that this paper was intended to cover only a part of the research program into the mechanism of heat transfer in insulation. Many points, including those made by the discussers, need clarification and I am confident that experimental research will clarify many of these problems.



**1531**

## SELECTION OF OUTSIDE DESIGN TEMPERATURE FOR HEAT LOAD ESTIMATION

By M. L. GHAI\* AND R. SUNDARAM\*\*, NEW DELHI, INDIA

**T**HE FIRST step in the air conditioning of a building is to estimate its heating or cooling load. Since this is an important basic step, several methods have been developed in the past few decades to improve the accuracy with which the heating or cooling load of a building can be predicted. This paper is devoted to the development of a rational, accurate, and yet simple method for the determination of the heat transmitted through barriers due to difference in outside and inside air temperatures.

This transmitted heat is generally estimated from the simple equation

$$q = UA (t_o - t_i) \quad (1)$$

where

$U$  = the coefficient of heat transmission.  
 $t_i$  = inside constant temperature, Fahrenheit.  
 $t_o$  = outside temperature, Fahrenheit.

It is evident that the outside temperature, or the outside design temperature as it is sometimes called, should be selected carefully so that Equation 1 will give the true heat transmission.

Before presenting the proposed method for selecting the outside temperature for use in Equation 1, it may be desirable to review briefly the basic factors which determine heat transfer through the building barriers.

### FACTORS DETERMINING MAXIMUM HEAT TRANSMISSION

Equation 1 for steady-state heat transfer is used in practice because of its simplicity. It does not represent the true condition because the actual heat transfer is non-steady due to varying outside temperature. As a result, the extent to which the outside temperature affects the cooling load of a barrier depends not only on the  $U$  factor (transmittance) of the barrier but also on the characteristics of the building materials.

Consider, for instance, two walls each having a  $U$  value of 0.42: (1) an 18-in. thick concrete wall and (2) a 1.26-in. thick wood wall. Both are exposed on the

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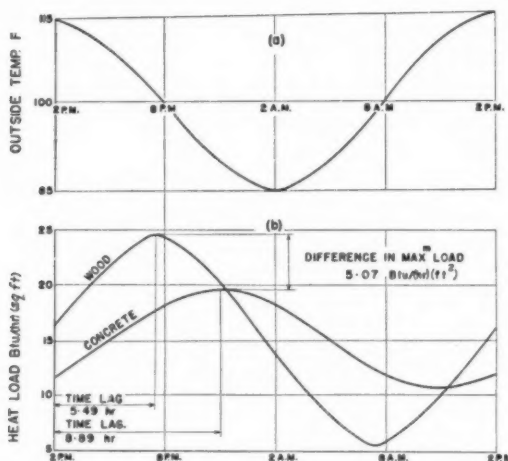


FIG. 1. (a) OUTSIDE TEMPERATURE VARIATION; (b) HEAT LOAD *vs.* TIME, FOR BARRIERS OF CONCRETE AND WOOD OF SAME COEFFICIENT OF HEAT TRANSMISSION, WITHOUT CONSIDERING THE AIR FILMS

outside to the temperature variation of Fig. 1, diagram (a), and on the inside to a constant temperature of 75 F. Fig. 2 gives the temperature distribution in the walls at three instants: when the outside temperature is maximum, when it is minimum, and when the heat flow to the inside is maximum. The slope of the temperature curves at the inner surface gives the heat flow to the inside, as shown in Fig. 1, diagram (b). The curves illustrate two well-known facts: (1) the maximum heat flow to the inside does not take place at the time when the outside temperature is maximum, but instead lags behind the outside temperature. The time lag for the concrete wall is larger. The outside maximum temperatures lasting for a short duration are, therefore, of less importance in the concrete wall and (2) the maximum heat transmittance is less for the concrete wall, even though the  $U$  value for the two walls and the outside weather conditions are the same. According to Equation 1, it would be concluded that two walls having the same  $U$  value and exposed to the same outside environment would have equal heat load.

It is thus necessary in developing a rational procedure for estimating heat load to take into account (1) the local weather conditions and (2) the characteristics of building materials and construction.

#### PRESENT METHODS

The basis of the recently used methods is summarized in the section on design outdoor conditions in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1954 as follows: *There are no hard and fast rules for selecting the design outdoor weather conditions to be used for a given locality or type of building or heating system, and the selection is to some extent a matter of judgment and experience.*

For estimating heating load, it has been recommended by the ASHAE Technical

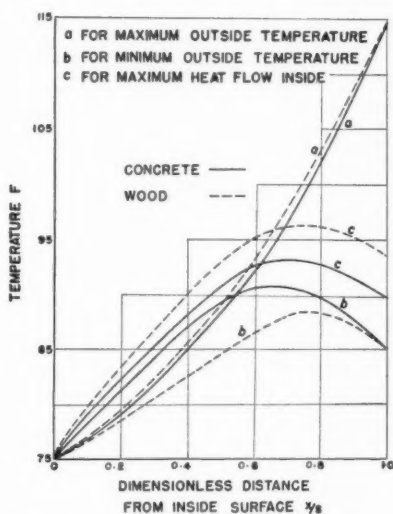


FIG. 2. TEMPERATURE DISTRIBUTION INSIDE BARRIERS OF CONCRETE AND WOOD HAVING SAME COEFFICIENT OF HEAT TRANSMISSION, FOR MAXIMUM AND MINIMUM CONDITIONS

Advisory Committee on Weather Design Conditions that the outside design temperature be the temperature equalled or exceeded during 97½ percent of the hours in December, January, February and March for the period of record. The recommendation for the cooling load estimate by the same committee is that the outside design temperature should be the maximum hourly temperature which has been equalled or exceeded 2½ percent of the total hours of June, July, August and September, for the period of record.

A common feature of the methods used at present is that the outside design temperature is fixed on the basis of the local weather data only, and consequently systems for all buildings in a particular locality are designed on the basis of the same outside temperature, irrespective of the variation in the characteristics of the building materials or construction. By excluding temperatures lasting for less than three hours or temperatures which have not been equalled or exceeded 2½ percent of the total hours, some consideration is given to the fact that the temperatures lasting for a short duration do not have their full effect on the building (since it takes time for the heat to penetrate the barrier). It is evident, however, that different buildings would be affected differently.

Analytical equations are available for precise calculation of the periodic heat flow through barriers. However, these equations involve elaborate and time-consuming mathematical calculations, and are rarely used by the engineer in the field. The authors have therefore tried to develop an equation having the requisite of simplicity and a reasonable amount of accuracy for computation of the correct heat transmission under periodic heat flow conditions.

## PROPOSED EQUATION FOR COOLING

If the outside design temperature,  $t_D$ , is defined as a fictitious constant temperature which produces the same amount of maximum heat transmission at the inner surface of the barrier as is produced by the actual varying outside temperature, the heat load can then be calculated from the simple steady-state calculated from the simple steady-state equation by substituting  $t_D$  in Equation 1 for the outside temperature

$$q = UA (t_D - t_i). \quad (2)$$

## NOMENCLATURE

$A$ = area of surface, square feet.	$t_{\min}$ = daily minimum outside temperature, Fahrenheit.
$c$ = specific heat, Btu per (pound) (Fahrenheit degree).	$t_o$ = outside temperature, Fahrenheit.
$k$ = thermal conductivity, Btu per (square foot) (hour) (Fahrenheit degree per foot).	$t_1$ = amplitude of daily temperature variation, Fahrenheit degrees.
$N_{Fo}$ = Fourier Number, $a\tau/s^2$	$U$ = overall heat transfer (air to air), Btu per (hour) (square foot) (Fahrenheit degree difference in temperature).
$N_{Nu}$ = Nusselt Number, $hs/k$	$x$ = distance from the inside surface of the barrier, feet.
$q$ = rate of flow, Btu per (hour) (square foot) (Fahrenheit degree).	$a$ = thermal diffusivity, square feet per hour.
$s$ = thickness of the barrier, feet.	$\rho$ = density, pounds per cubic foot.
$t$ = temperature, Fahrenheit.	$\tau$ = time, hours.
$t_D$ = equivalent outdoor design temperature, Fahrenheit.	$\tau_e$ = time needed for one full cycle, hours.
$t_i$ = inside constant temperature, Fahrenheit.	$\Omega$ = decrement factor for amplitude of temperature variation.
$t_m$ = daily mean outside temperature, Fahrenheit.	
$t_{\max}$ = daily maximum outside temperature, Fahrenheit.	

Adopting this definition of  $t_D$ , it is shown in Appendixes A and B that the outside design temperature,  $t_D$ , may be calculated from the following equation:

$$(t_{\max} - t_D)/(t_D - t_m) = F \quad (3)$$

where

$$F = 0.0007 X^2 (1 + 2Y\sqrt{Y})$$

$$X = \rho cs$$

$$Y = (1/k)s$$

$$t_{\max} = \text{daily maximum outside temperature, Fahrenheit.}$$

$$t_m = \text{daily mean outside temperature, Fahrenheit.}$$

$$k = \text{thermal conductivity, Btu per (square foot) (hour) (Fahrenheit degree per foot).}$$

$$\rho = \text{density, pounds per cubic foot.}$$

$$c = \text{specific heat, Btu per (pound) (Fahrenheit degree).}$$

$$s = \text{thickness of the barrier, feet.}$$

The left-hand side of the equation depends on the outside temperature variation—that is, on the local weather conditions. The right-hand side of the equation represents the effect of the construction and the materials—that is, characteristics of the particular building under consideration.

For a very thick barrier, the right-hand side approaches infinity, where  $t_D = t_m$ . The design temperature of a very thick barrier should, therefore, be taken equal to the daily mean temperature. For a thin barrier, the right-hand side approaches

zero, where  $t_D = t_{\max}$ . The design temperature for a very thin barrier should, therefore, be taken equal to the daily maximum temperature. In any particular locality with given  $t_{\max}$  and  $t_m$ , when the thickness, density, or specific heat of the barrier increases and the thermal conductivity decreases, the right-hand factor becomes larger, and the design temperature becomes smaller. The daily mean temperature is the lower limiting value.

The calculation of the right-hand side of the equation requires data on the properties of construction materials. Such data can be obtained from any appropriate hand-book. The left-hand side requires the typically high daily maximum and daily mean temperatures to be expected in the locality under consideration.

To illustrate the use of the proposed equation, let it be required to determine the outside design temperature for the 18-in. thick concrete barrier and 1.26-in. thick wood barrier (mentioned earlier in this paper) in a locality where the daily maximum and mean temperatures are 115 F and 100 F, respectively. Further, let it be required to calculate the maximum cooling loads for the two barriers, the overall coefficient of heat transmission for both the barriers being 0.42 Btu per (hr) (sq ft) (F deg), and the inside temperature being 75 F.

For the concrete wall

$$\rho c = 30, 1/k = 1, \text{ and}$$

$$s = 18/12 = 1.5$$

$$X = 30 \times 1.5 = 45$$

$$Y = 1 \times 1.5 = 1.5$$

$$F = 0.0007 X^2 (1 + 2Y\sqrt{Y}) = 6.59$$

$$(t_{\max} - t_D)/(t_D - t_m) = (115 - t_D)/(t_D - 100) = 6.59,$$

giving

$$t_D = 101.98 \text{ F}$$

$$\text{Maximum heat transfer} = 0.42(101.98 - 75) = 11.3 \text{ Btu per (hr) (sq ft)}$$

For the wood wall

$$\rho c = 17.5, 1/k = 14.3, \text{ and } s = 1.26/12 = 0.11$$

$$X = 17.5 \times 0.11 = 1.89;$$

$$Y = 14.3 \times 0.11 = 1.5$$

$$F = 0.0007 X^2 (1 + 2Y\sqrt{Y}) = 0.01$$

$$(t_{\max} - t_D)/(t_D - t_m) = (115 - t_D)/(t_D - 100) = 0.01,$$

giving

$$t_D = 114.85 \text{ F}$$

$$\text{Maximum heat transfer} = 0.42(114.85 - 75) = 16.7 \text{ Btu per (hr) (sq ft)}$$

The maximum heat load for the wood wall is about 48 percent higher than that of the concrete wall, although the coefficient of heat transmission is the same. These walls are of course somewhat extreme cases of construction.

#### PROPOSED EQUATION FOR HEATING

Equation 3 for cooling can also be applied to heating of buildings, after slight modification. For heating, the equation becomes:

$$\frac{t_{\min} - t_D}{t_D - t_m} = 0.0007 X^2 (1 + 2Y\sqrt{Y}) \dots \dots \dots (4)$$

Only the  $t_{\max}$  of the equation for cooling is replaced by  $t_{\min}$  when it is used for heating. It is quite evident that for cooling the maximum outside temperature,  $t_{\max}$ , is important while for heating the minimum outside temperature,  $t_{\min}$ , is important. In case of heating, the design temperature,  $t_D$ , lies between the daily minimum and the daily mean temperature. For thick structures of heavy materials  $t_D$  is close to  $t_m$ , while for thin structures of light materials  $t_D$  approaches  $t_{\min}$ .

To illustrate, consider that the walls of wood and concrete used in the previous examples are situated in a locality where the daily mean and minimum temperatures are 0 F and -15 F respectively.

For the concrete wall,

$$(t_{\min} - t_D)/(t_D - t_m) = (-15 - t_D)/(t_D - 0) = 6.59$$

hence

$$t_D = -1.98 \text{ F}$$

$$\text{Maximum heat transfer} = 0.42(-1.98 - 75) = -32.3 \text{ Btu per (hr) (sq ft)}$$

For the wood wall,

$$(t_{\min} - t_D)/(t_D - t_m) = (-15 - t_D)/(t_D - 0) = 0.01$$

hence

$$t_D = -14.25 \text{ F}$$

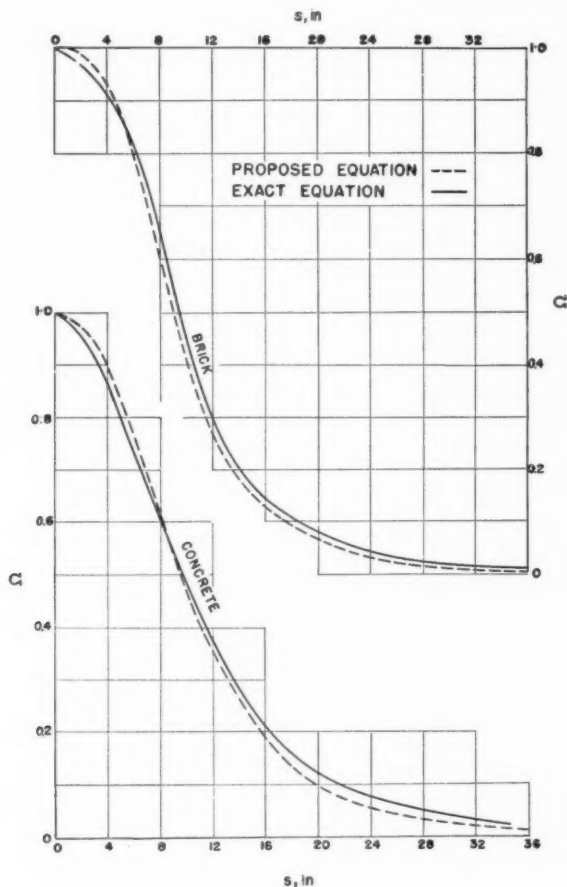
$$\text{Maximum heat transfer} = 0.42(-14.85 - 75) = -37.7 \text{ Btu per (hr) (sq ft)}$$

The proposed method offers another advantage which is likely to make it convenient and desirable. One reason why none of the methods for determining design temperature has been universally accepted in practice is that they require detailed weather data, collection of which is expensive and time consuming. For instance, if the design temperature for a locality is to be determined from hourly records, it is necessary to have a complete hourly temperature record for every day during a period of 10 years. The method proposed in this paper does not require the hourly temperatures; it requires only the daily maximum and minimum temperatures which are relatively easy to obtain.

Although the proposed method overcomes the difficulty of collecting hourly temperatures, it still incorporates a factor which must be fixed somewhat arbitrarily by the individual designer until a standardized procedure is established. It is necessary to decide whether the daily maximum and mean temperatures are to be taken as the maximum in a year, or as an average of such temperatures in 5, 10, or 20 years.

#### ASSUMPTIONS AND ACCURACY

The proposed equations for heating and cooling are based on the commonly used film coefficients, 1.65 for inside and 4.0 for outside. The actual outside temperature variation has been taken as a single sinusoidal function of time. Also, it may be pointed out that these equations give a basis for selection of outside design temperature making allowance for the periodic variation of outside temperature, and are to be used for the calculation of transmission heat losses only. The heat load due to solar radiation, and other causes should be estimated separately.


 FIG. 3.  $\Omega$  vs.  $s$ , FOR CONCRETE AND BRICK

The accuracy of the proposed Equations 3 and 4 has been indicated in Figs. 3 and 4. The effect of the building construction and materials is represented by a factor,  $\Omega$ , which is given by,

$$\Omega = \frac{1}{1 + 0.0007 X^2 (1 + Y\sqrt{Y})}$$

or

$$\Omega = \frac{t_D - t_m}{t_{\max} - t_m}$$

or

In Figs. 3 and 4, values of  $\Omega$  obtained from the proposed equations have been compared with the values obtained from exact mathematical equations (for details see Appendixes A and B). The comparison has been made for barriers of various thicknesses and different commonly used materials (concrete, brick, wood, cork). For thick heavy walls,  $\Omega$  approaches zero; the design temperature should be close to the daily mean temperature. For thin light walls,  $\Omega$  approaches one; the design temperature should be close to the daily maximum temperature in summer or daily minimum temperature in winter. Another interesting feature of the curves is their shape. As  $s$  increases,  $\Omega$  at first decreases gradually, then rapidly, and then again slowly. This indicates the economical thickness of the barriers. For instance, in case of brick walls, when  $s$  increases from 4 in. to 16 in.  $\Omega$  decreases from 0.91 to 0.15, *i.e.*, by 0.76; but when  $s$  increases from 16 to 28 in.  $\Omega$  decreases from 0.15 to 0.02, *i.e.*, by only 0.13. The curves indicate that an increase of thickness beyond about 20 in. for concrete, 16 in. for brick, 8 in. for wood, and 10 in. for cork, obtains much smaller returns in the form of savings in heat transfer than corresponding increases of thickness for thinner walls.

The present paper deals only with homogeneous constructions. The application of the proposed method to multilayer constructions is being considered.

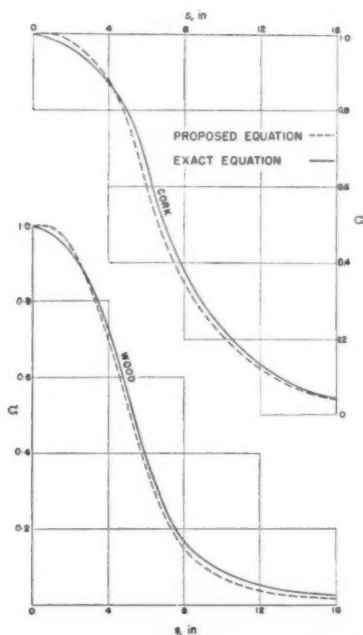


FIG. 4.  $\Omega$  vs.  $s$ , FOR WOOD AND CORK

# APPENDIX A

## DERIVATION OF THE EQUATION FOR OUTSIDE DESIGN TEMPERATURE WITHOUT CONSIDERING THE AIR FILMS

Let the outside temperature,  $t_o$ , vary periodically according to the equation:

$$t_o = t_m + t_1 \cos [2\pi (\tau/\tau_c)] \quad \dots \quad (A-1)$$

where  $t_m$  and  $t_1$  are the daily mean temperature, and the amplitude of the daily temperature variation respectively.

The analysis is carried out for a single sinusoidal function only, recognizing that, for all practical purposes, the actual daily temperature variation may be assumed to be close to a single sinusoidal function. For precise results, exact analysis may be made by expressing the daily temperature variation as a Fourier series, using methods of harmonic analysis, and treating separately each term of the series in a similar manner.

The solution of the Fourier-Poisson's equation,

$$\partial t / \partial \tau = a (\partial^2 t / \partial x^2),$$

gives the temperature distribution and the temperature gradient inside the barrier. The maximum value of the temperature gradient at the inner surface may be expressed in terms of the Fourier number,  $N_{Fo}$ .

$$\frac{t_m - t_i}{s} + 2\sqrt{2} \lambda_1 B t_1 \quad \dots \quad (A-2)$$

where

$$\lambda_1 = \sqrt{\pi / \alpha \tau_c}$$

$$B = \frac{1}{\sqrt{2(\cosh 2\sigma - \cos 2\sigma)}}$$

$$\sigma = \sqrt{\frac{\pi}{N_{Fo}}}$$

and

$$N_{Fo} = \frac{\alpha \tau_c}{s^2}$$

Defining the outside design temperature  $t_D$  as a fictitious constant outside temperature which produces the same amount of maximum heat transmission at the inner surface as is produced by the actual varying outside temperature, Equation A-2 becomes

$$\frac{t_D - t_i}{s} = \frac{t_m - t_i}{s} + 2\sqrt{2} \lambda_1 B t_1$$

or

$$t_D = t_m + 2\sqrt{2} \lambda_1 s B t_1$$

or

$$t_D = t_m + 2\sqrt{2} \sigma B t_1$$

This may also be expressed in the form

$$t_D = t_m + \Omega t_1 \quad \dots \quad (A-3)$$

where

$$\begin{aligned} \Omega &= 2\sqrt{2} \sigma B \\ &= \sqrt{\frac{4\pi}{N_{Fo}}} / \sqrt{\cosh \sqrt{\frac{4\pi}{N_{Fo}}} - \cos \sqrt{\frac{4\pi}{N_{Fo}}}} \end{aligned}$$

Equation A-3 clearly indicates that the outside design temperature should be selected by considering the outside temperature variation represented by  $t_{\max}$  and  $t_m$  as well as the factor  $\Omega$ , which depends on the building characteristics. Expressing  $\cosh$  and  $\cos$  terms as infinite series, and simplifying,  $\Omega$  may be expressed as

$$\Omega = \frac{1}{\left[1 + 2 \frac{(2\sigma)^4}{6!} + \dots\right]^{\frac{1}{2}}}$$

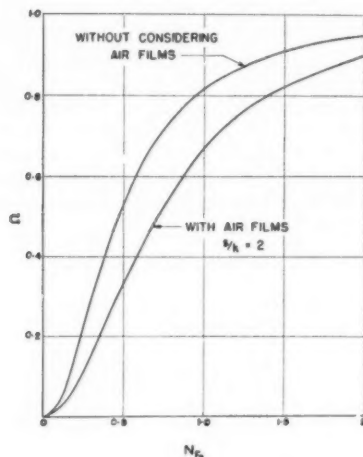


FIG. A-1. COMPARISON OF VARIATION OF THE DIMENSIONLESS QUANTITY  $\Omega$  WITH THE FOURIER NUMBER,  $N_{Fo}$ , FOR BARRIERS WITH AIR FILMS AND  $(s/k) = 2$ , AND BARRIERS WITHOUT AIR FILMS

Since  $(2\sigma)^4/6!$  is generally less than 1, powers of  $2\sigma$  higher than the fourth may be neglected.

Therefore,

$$\Omega = \frac{1}{1 + \frac{(2\sigma)^4}{6!}}$$

Expressing  $\sigma$  in terms of the Fourier number,

$$\Omega = 1/[1 + (0.22/N_{Fo}^2)] \quad \dots \quad (A-4)$$

Thus  $\Omega$  is a function of the well-known Fourier number. Fig. A-1 gives a plot of  $\Omega$  vs.  $N_{Fo}$  (with and without considering the air films). The curves rise rapidly for the lower values of  $N_{Fo}$  and rise slowly for the higher values of  $N_{Fo}$ .

## APPENDIX B

## DERIVATION OF EQUATION FOR OUTSIDE DESIGN TEMPERATURE CONSIDERING AIR FILMS

Analytical equations for periodic heat-flow through homogeneous walls considering the outside as well as inside air-film coefficients have been obtained by Alford, Ryan and Urban,<sup>1</sup> and have been further checked by Mackey and Wright.<sup>2, 3</sup> Applying their equations, and assuming the generally acceptable values of the inside and outside film coefficients of 1.65 and 4.0 Btu per (hr) (sq ft) (F deg) respectively, the following equation for the maximum heat load may be derived,

$$\text{Maximum heat load} = \frac{t_m - t_i}{\frac{1}{1.65} + \frac{s}{k} + \frac{1}{4.0}} + 1.65 t_i \quad \dots \quad (\text{B-1})$$

where

$$\begin{aligned} \lambda &= \frac{5.65 k \lambda_1}{\sqrt{f^2 + g^2}} \\ f &= 5.65 k \lambda_1 (\cos \lambda_1 s \cosh \lambda_1 s + \sin \lambda_1 s \sinh \lambda_1 s) + 6.6 \sin \lambda_1 s \cosh \lambda_1 s + 2k^2 \lambda_1^2 \cos \lambda_1 s \sinh \lambda_1 s \\ g &= 5.65 k \lambda_1 (\cos \lambda_1 s \cosh \lambda_1 s - \sin \lambda_1 s \sinh \lambda_1 s) + 6.6 (\cos \lambda_1 s \sinh \lambda_1 s - 2k^2 \lambda_1^2 \sin \lambda_1 s \cosh \lambda_1 s) \end{aligned}$$

$$\lambda = \sqrt{\frac{\pi}{\alpha \tau_c}}$$

According to the definition of the design temperature given in Appendix A,

$$\text{Maximum heat transfer} = [t_D - t_i] / [(1/1.65) + (s/k) + (1/4.0)]$$

therefore,

$$t_D = t_m + 1.65 [(1/1.65) + (s/k) + (1/4.0)] \lambda t_i$$

or

$$t_D = t_m + \Omega t_i$$

where

$$\Omega = 1.65 [(1/1.65) + (s/k) + (1/4.0)] \lambda \quad \dots \quad (\text{B-2})$$

These equations may be used for determination of exact values. However, the unwieldy nature of  $\Omega$  indicates the necessity of a simpler equation. To obtain a simple equation, the following procedure was adopted.

The present problem remained a problem of pure conduction so long as the air films were not considered. Due to the introduction of air films it becomes one of conduction with convection involving both the Fourier and the Nusselt number. Therefore, let  $\Omega$  be represented by

$$\Omega = \phi (N_{Fo}, N_{Nu}) = \phi [(a\tau_c/s^2), [h_i (s/k)], [h_o (s/k)]]$$

where  $[h_i (s/k)]$  and  $[h_o (s/k)]$  are the Nusselt numbers corresponding to the inside and outside air film coefficients, and  $a\tau_c/s^2$  is the Fourier number. The thickness of the wall,  $s$ , is taken as the characteristic length in the Nusselt numbers.

If the constant values of the inside and outside film coefficients are substituted,  $\Omega$  may be represented simply by,

$$\Omega = \phi [(a\tau_c/s^2), (s/k)]$$

$s/k$  must have such functional relationship that  $\Omega$  should be unity for  $s/k = 0$ , and zero for  $s/k = \infty$ . This is indicated by the equation  $t_D = t_m + \Omega t_i$ .

For walls of negligible thickness [ $(s/k)$  approaching 0],  $t_D$  should evidently be  $t_{max}$  or  $t_m + t_i$ , that is,  $\Omega$  should be unity. Similarly for walls of infinitely large thickness [ $(s/k)$  approaching  $\infty$ ],  $t_D$  should evidently be equal to  $t_m$ , or  $\Omega$  should be zero.

In order to see how the presence of air films affects the relationship between  $\Omega$  and  $N_{Fo}$ , the exact relation between  $\Omega$  and  $N_{Fo}$  for walls of  $s/k = 2$  was determined and

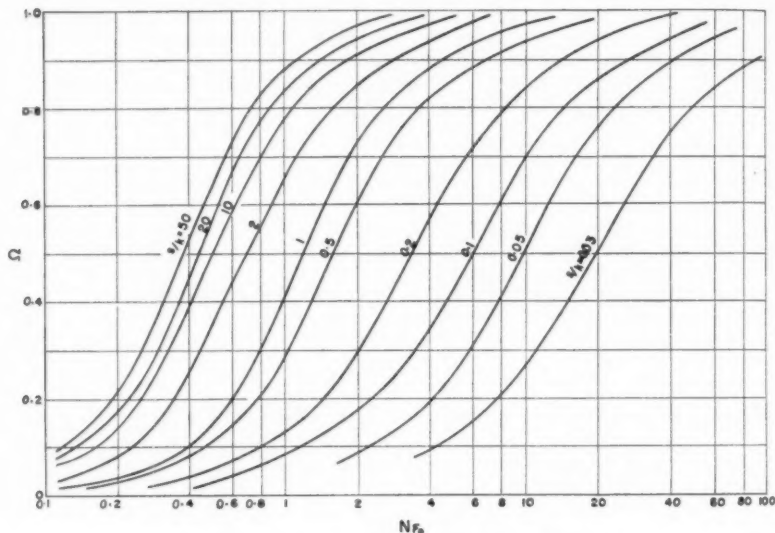


FIG. B-1.  $\Omega$  vs.  $N_{Fo}$  FOR BARRIERS WITH AIR FILMS

plotted for walls with as well as without air films. The curves were of similar shape for both conditions. The method of analysis which follows was then tried.

A number of sections of the surface

$$\Omega = \phi [(a\tau_c/s^2), (s/k)]$$

by the planes  $s/k = \text{constant}$  were obtained as shown in Fig. B-1. All of these indicated similarity with the plot of  $\Omega$  vs.  $N_{Fo}$  without considering the air films, shown in Fig. A-1. Thus  $\Omega$  could be of the form

$$\Omega = \frac{1}{1 + \frac{0.22}{N_{Fo}^2} + \left(\frac{s}{k}\right)} \quad \dots \quad (B-3)$$

Another form of  $\Omega$  satisfying the end-conditions is given by

$$\Omega = \left[ \frac{1}{1 + \frac{0.22}{N_{Fo}^2}} \right] \left[ \frac{1}{1 + f\left(\frac{s}{k}\right)} \right]$$

This was however rejected as being too complicated.

To arrive at the nature of  $f(s/k)$ , Equation B-3 was written as

$$\frac{1-\Omega}{\Omega} = \frac{0.22}{N_{F0}} f\left(\frac{s}{k}\right)$$

$$\text{or } \log \frac{1-\Omega}{\Omega} = \log 0.22 f\left(\frac{s}{k}\right) - 2 \log N_{F0}$$

Fig. B-2 shows  $(1-\Omega)/\Omega$  vs.  $N_{F0}$ . The points shown in Fig. B-2 are the points, selected somewhat at random, for which values were calculated. These points did not lie exactly on straight lines  $(1-\Omega)/\Omega$  vs.  $N_{F0}$  having slope of 2 as assumed in the above equation. However, lines having slope of 2 could be fitted with a fair degree of accuracy. These lines of slope 2 were used to determine  $f(s/k)$  in terms of  $(s/k)$ . Values of  $f(s/k)$ , thus obtained, were plotted against  $(s/k)$  in Fig. B-3. In the lower region, for  $0.03 < s/k < 0.2$ , the plot of  $f(s/k)$  could be approximated to a straight line from which the following relationship was obtained:

$$f\left(\frac{s}{k}\right) = \frac{1.83}{\left(\frac{s}{k}\right)^2} \text{ for } 0.03 < \frac{s}{k} < 0.2 \quad \text{. . . . . (B-4)}$$

In the higher region, for  $0.2 < s/k < 60$ , the following approximate relationship was obtained:

$$f\left(\frac{s}{k}\right) = \frac{1.83}{\left(\frac{s}{k}\right)^2} + 2 \frac{1.83}{\left(\frac{s}{k}\right)^3} \quad \text{. . . . . (B-5)}$$

In arriving at Equations B-4 and B-5, fractional powers of  $s/k$ , with the exception of square-root, were avoided since these were likely to make the equation difficult. The range  $0.2 < s/k < 60$  covers most of the commonly used building constructions. It was therefore considered desirable that, for the sake of simplicity,  $f(s/k)$ , valid in the

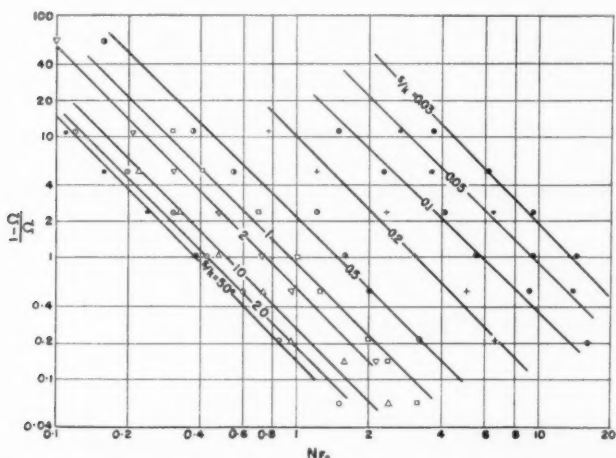
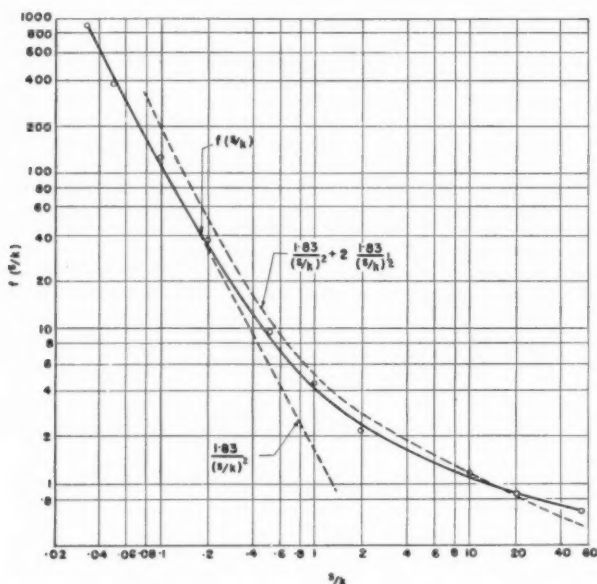


FIG. B-2.  $[(1-\Omega)/\Omega]$  vs.  $N_{F0}$  FOR BARRIERS WITH AIR FILMS

FIG. B-3.  $f(s/k)$  vs.  $s/k$ , FOR BARRIERS WITH AIR FILMS

higher range, might be considered generally applicable for all practical purposes. Fig. B-3 further indicated that the use of Equation B-5 in the lower range of  $s/k$  did not create abnormally high errors.

Substituting Equation B-5 in Equation B-3 the equation for  $\Omega$  becomes

$$\Omega = \frac{1}{1 + \frac{0.22}{N_{Fe}} \left[ \frac{1.83}{\left(\frac{s}{k}\right)^2} + 2 \frac{1.83}{\left(\frac{s}{k}\right)^{1/2}} \right]} \quad \text{. . . . . (B-6)}$$

Substituting for  $N_{Fe}$ , and simplifying, Equation B-6 becomes

$$\Omega = \frac{1}{1 + 0.0007 \rho^2 c^2 s^2 \left[ 1 + 2 \left(\frac{s}{k}\right) \sqrt{\frac{s}{k}} \right]} \quad \text{. . . . . (B-7)}$$

Thus, the equation for estimating  $t_D$  is

$$t_D = t_m + \Omega t_i$$

where

$$\begin{aligned} \Omega &= 1/(1 + F) \\ F &= 0.0007 X^2 (1 + 2Y\sqrt{Y}) \\ X &= \rho c s \\ Y &= (1/k)s \end{aligned}$$

$$\text{Since } t_D = t_m + [1/(1 + F)] t_1 \\ F = (t_m - t_D + t_1)/(t_D - t_m)$$

$$\text{Since } (t_m + t_1) = t_{\max}$$

this equation may be written

$$(t_{\max} - t_D)/(t_D - t_m) = F \dots \dots \dots (B-8)$$

which is another expression for obtaining  $t_D$ .

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2. Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 49, 1943, pp. 148-74).
3. ASHVE RESEARCH REPORT No. 1255—Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 50, 1944, pp. 293-312).

#### DISCUSSION

CARL W. SIGNOR, Detroit, Mich., (WRITTEN): The authors should be commended on their analysis of the behavior of the heat flow through building walls with a varying outdoor temperature.

This paper makes it very apparent that the heat transmission co-efficients as given in the GUIDE are on the safe side but not always correct for thick walls. The equation as set up by the authors will be exceedingly helpful when more exact figures on heavy walls are desired.

One condition that I have noticed in the heating of heavy walled buildings in the Detroit area is that when three successive days of near 0 F temperature occur, the building heating requirements increase each successive day. Then during the first day that the outdoor temperature rises to normal winter conditions, the building will still use an excessive amount of steam for heating. We know this behavior is due to the heat storage effect of the walls, but it does show the time delay of the heat flow through the walls as pointed out in the paper.

A. I. BROWN, Columbus, Ohio (WRITTEN): The authors have made an interesting mathematical analysis of the complicated problem of determining the equivalent outside design temperature for heat load estimation when consideration is given to local weather conditions, the time lag for walls which have different rates of heat storage capacity, and the resulting differences in maximum heat flow which occur for such walls even though they may have similar values of the overall coefficient  $U$ .

The title implies that the paper deals specifically with heating load calculations. In the paper, however, the authors apply their proposed equation to cooling load as well as to heating load estimation.

The authors have made it clear that their proposed equations are to be used only for the calculation of transmission heat losses (or gains), and that loads due to solar radiation and other causes should be estimated separately.

With this limitation in mind, the reader may observe that any calculation of an outside design temperature for transmission gains, as applied to cooling problems, can have very limited application since it is independent of the heat gains due to solar radiation. It appears then that this method when used in conjunction with the Guide table of *Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Walls*, would apply only to the north or shaded walls. Furthermore, it may be

noted that the result given by the authors' equations is the *maximum* heat gain by transmission; and since this maximum heat flow through shaded walls usually does not occur at the same hour as the maximum total heat gain, which includes the gain through glass and roofs, and other sources its application is further restricted.

In the case of calculations of heating loads, where heretofore no account has been taken of unsteady state heat transfer, the authors' method provides a refinement to calculations which may well be justifiable in the case of transmission through thick masonry walls, especially in localities where cold spells are of short duration and consequently where an appreciable difference may exist between the minimum outside temperature and the minimum daily mean temperature.

The sample calculations included in the authors' paper, which are based upon a daily mean of 0 F and a minimum of -15 F, show a significant difference between the calculated equivalent outside design temperature and the minimum outside temperature in an 18-in. concrete wall—a difference of 13.02 degrees. A similar calculation for a 12-in. brick wall would show a difference of 11.2 degrees. In the case of lighter construction, such as frame walls and windows, however, the difference in temperature appears to be too small to justify such a calculation. This means, then, that users of the authors' method would have to apply different outdoor design temperatures for heavy masonry walls than for air-infiltration calculations and for windows and possibly other surfaces of the same building.

In spite of the foregoing limitations to the use of the authors' method, they are to be commended for showing so clearly that under conditions of fluctuating temperature, walls having similar *U* values may be quite different in their effectiveness as heat barriers. Their paper shows also that the reduction in heating and cooling requirements for buildings with heavy masonry walls, as compared with walls of lighter construction, is due not only to the fact that, because of time lag, the peak loads for walls, roofs and windows do not occur at the same hour, but the reduction in requirements is due also to a smaller temperature differential across the heavy walls at the time of the maximum heat flow.

F. W. MCGHAN, Washington†, D. C. (WRITTEN): The basic concept for weighting the heat transmission through building sections due to the thermal lag as related to the density and specific heat of materials appears to be a valid contribution in the approach toward the establishment of more factual transmission coefficients. Considering present methods of testing materials to determine their thermal conductivity, the change in moisture content as actually installed may also be a factor to be considered.

The method of determining an equivalent outside design temperature, and its use in actual transmission calculation would have limited usefulness. The authors state that the walls used for illustration purposes are *somewhat extreme cases of construction*. It is suspected that if the method of calculation indicated (which shows a variation of 5.4 Btu per sq. ft. for the extreme conditions used) were applied to components normally used in residential construction, the magnitude of the variation would lose most of its significance. In addition the proposed method would require the use of a large number of equivalent outside design temperatures for a single structure—one for each different type of structural section used—which would appear impractical.

The following items may complicate application of the proposed method to actual residential structures:

1. Reflective type insulations (density and specific heat as related to attributed thermal value).
2. Solar effect on exterior walls and relative of elevated surface temperatures on thermal gradient and rate of transmission.
3. Construction using non-homogeneous materials, particularly those having air spaces and involving different directions of heat flow.

Present methods of calculating residential heat gains (24 hour method) recognize the

† Federal Housing Administration.

transmission lag and heat storage effect of materials in the development of simplified factors. These items might well be incorporated in our current methods of heat loss computation to reduce the difference now existing between measured and calculated results.

JOHN EVERETTS, JR., Philadelphia, Pa., (WRITTEN): The problem of selecting outside design temperatures for heating and cooling load estimates is not a simple one as attested by the many papers and discussions now in the literature.

The authors, in the paper just presented, have attempted, by analytical methods, to arrive at an equivalent temperature which considers structures of different densities and specific heat but with the same over-all coefficient of heat transfer. The effect of solar radiation has not been included.

In reviewing this paper two questions came to mind; (1) how close does the heat transfer calculated by this method approach the actual measured heat transfer and (2) how important is the deviation in end results to the total calculated load.

In discussing the first question, a calculation was made following the sample calculations in the paper but with values as follows:

$t_{max}$	95 F.
$t_m$	80 F.
Room Temp.	75 F.
Wall 8-in brick with $\frac{3}{4}$ -in metal lath and plaster	
Density	100
Specific heat	0.20
K	8.0
1/K	1.25
U	0.46
S	0.73

Substituting these values, the equivalent temperature was calculated to be approximately 90.8 F. This results in a heat transfer of 7.3 Btu per sq. ft. per hr.

If we refer to Laboratory Report No. 1195 on Heat Gain Thru Walls, published in Vol. 48 of the ASHVE Transactions, dated 1942, curve 1 of Fig. 1, page 95, shows the heat flow for a north exposed wall of 8-in brick construction, and with no solar radiation, to reach a maximum of 4.4 Btu per sq. ft. per hr. This would indicate a calculated transfer of approximately 66 per cent more heat than the actual measured value. This difference may be due to—

1. Storage effect in mass of structure.
2. Reverse flow of heat on decline of temperature after maximum has been reached.
3. Use of incorrect values in formula as compared to test wall.

The second question as to the importance of the difference in transfer to any total load may be illustrated by a typical example.

A suburban department store consisting of basement, first floor and second floor has 60,000 sq. ft. of roof area, 25,600 sq. ft. of wall and glass area, of which 13.7 percent or 3,500 sq. ft. is glass. This is a very low glass/wall ratio and should tend to favor the results of this problem. The load breakdown is as follows:

	Btu/hr.	Percent Total Load
Total Load	7,400,000	100
Transmission		
Walls	62,000	0.84
Roof	175,000	2.35
Glass	60,000	0.81
Sub Total	297,000	4.00
Solar Radiation		
Walls	—	—
Roof	265,000	3.58
Glass	134,000	1.81
Sub Total	399,000	5.39
People, Lights, Ventilation	6,704,000	90.61

From these figures, it can readily be seen that with a wall transmission of only 0.84 percent of the total load a wide variation in design temperature or wall construction would have little effect on the over-all design of an air conditioning system. The roof transmission is important insofar as the air quantity for the top floor is concerned. However, even a wide deviation (48 percent as shown in the authors' example) of this transmission would not seriously affect the air conditioning design because of the order of magnitude of the solar heat gain and gain from people and lights.

The analytical method presented by the authors is of academic interest as far as they go and if they could extend their method to include solar radiation and storage, then the value of this method would be greatly enhanced.

S. A. HEIDER, Washington, D. C., (WRITTEN): *Average Temperatures as Design Data*—This paper points out that minimum and maximum weather temperature data will be needed in suitable form for use with the method. A decision between using the highest temperature values for a year or averages based on several years is left open.

The highest temperature values, in my opinion, are too extreme. Average values on the other hand mean little to designers generally. Most designers, I believe, are not satisfied with averages, but prefer to select design temperatures to suit actual conditions—a little higher outside temperature for a house located on hot city streets, a little lower temperature for a building in the suburbs.

Rather than average temperatures, I feel that weather data should be given in terms of *incidence of occurrence* for a series of temperatures, i.e., the number of times selected temperatures have been equaled or exceeded over a period of record. This permits the designer to select the temperature best suited to his particular conditions. Moreover, it permits him to tell his client just how many days during the cooling season the system can be expected to meet the load.

*Priority of Weather Data*—The method proposed in this paper requires both maximum and mean daily temperatures. The values for these temperatures ought to be obtained from simultaneous readings of record in order that they may keep their proper relationship in the formula. This requires data from simultaneous maximum and minimum daily readings and involves considerable preparation, presumably by the Weather Bureau.

Since fresh air is an important item in determining the air conditioning load, it seems to me that wet bulb data is needed even more than maximum and minimum dry bulb data. In other words, before we devote too much attention to data for simultaneous maximum and minimum dry bulb, we ought first to do something about preparing data for simultaneous maximum dry bulb and wet bulb.

H. T. GILKEY, Urbana, Ill., (WRITTEN): This paper presents a very promising method for determining the maximum heat flow by conduction through a homogeneous wall. In addition, it presents what may prove to be the basis for a reevaluation of the outdoor design temperatures for cooling in any locality. It, like other work in the field, is not the final answer to all of the problems associated with the calculation of cooling loads.

As the authors point out, the work must be extended to include non-homogeneous walls before it will have wide application. It is hoped that this extension of the work will include the types of wall construction commonly used in residences. Once this is done it may be possible to substitute what the authors have termed as  $t_D$  for what we now call outdoor design temperature. It seems likely, however, that in a given locality several values of  $t_D$  can be determined depending upon the type of construction under consideration. This situation of multiple listings would not be desirable, and before the concept presented can be of practical use it will be necessary to find some method of eliminating the necessity for listing more than one design value for a given locality.

Another limitation which the authors point out is the absence of radiation effects in the equation. The effect of solar radiation on heat gains through walls is two-fold. In the first place, it has the same effect as an increased outdoor air temperature. Unfortunately, this effect is not uniform over all surfaces of a building, but it is appreciable

even though a wall may not be exposed directly to the sun. In addition, there are appreciable radiation effects on the interior surfaces of exterior walls. These usually occur because of re-radiation and reflection from interior surfaces exposed to radiation through glass areas.

Even though it may be very desirable to be able to calculate the maximum heat flow through a given wall, it is very unlikely that the maximum heat flow will occur simultaneously for all exposed surfaces of a structure. For instance, the heat flow through an east wall will almost without exception reach its maximum much earlier than does the heat flow through a west wall. Consequently, being able to determine only the maximum heat flow will not be sufficient for an accurate calculation of the heat gain to a building.

Some of these comments apply equally to cooling and to heating. It may well turn out, however, that the concept of  $t_D$  will prove much more valuable as an aid in estimating heat loss than in estimating heat gain, since heat losses during periods of maximum load are not subject to solar effects.

E. P. PALMATIER, Syracuse, N. Y.: I think the author is to be congratulated for offering one means of simplifying a very complex problem. We have a field day here for discussers because this is such a complex problem that it is very easy to find other avenues of attack.

I think the point has been adequately covered by the preceding discussers that the transmission component of the load in many types of structures, particularly heavy structures, fades into relative insignificance because of the large light loads, people loads and ventilation loads that are normally found in commercial structures.

It is important to point out that even in very light structures it has been shown before the Society that a much simpler method of determining the heat gain is available other than this business of trying to determine the periodic heat flow. That is by integrating the various components of the load over a long period of time.

One other point I would like to make is the fact that there is at least as much material within these structures as there is in the wall, and consequently the storage effects that come about through the presence of all this material inside the structure, that is inside the outer skin, is as important a problem as the periodic heat flow through the wall.

P. R. ACHENBACH, Washington, D. C.: I want to compliment the authors for this first step, at least, in simplifying periodic heat flow through walls.

There are a number of concepts being advanced nowadays about improving and simplifying methods for computing heat gain or heat loss. One of those is related to the frequency of recurrence of low or high outside values, as the case might be. Another relates to adjusting the design temperature for the diffusivity of the wall and this paper this morning is related to the latter idea.

I believe there is sound theoretical ground for using different outside design temperatures for walls of different diffusivity, and that is what this factor,  $F$ , in this paper is related to.

However, before we list a lot of different values of design temperature, there should be adequate guidance for the user in when he should use what value.

We have two values of outside design temperature for winter use in the GUIDE now and there are some questions as to which and when to use them.

There is a good possibility that there will be more than two values in some cases. In fact, I believe the 1955 GUIDE will list four values for Canadian cities. Unless we can provide adequate guidance for the user as to what determines the value to choose, there will be more confusion than now exists.

The value,  $F$ , that is used in this paper could be tabulated for various kinds of walls and the maximum or minimum temperature, and the daily amplitude could also be used, which would serve as tools for putting this formula into use.

We know that there is a lot of confusion and disagreement among engineers and contractors as to what is the heat loss or heat gain of a building, and actually it is

difficult to promote a fair competition among individuals when there are different methods of computation.

I believe this idea that has been advanced here is worthwhile and should be considered by those who are proposing changes in the method of selecting design temperatures and changes in the methods of computing heat loss and heat gain. The limitations of the method described, I think, have been adequately covered by other commenters.

**AUTHORS' CLOSURE (Doctor M. L. Ghai):** The authors appreciate the views expressed by the discussers, especially since the paper concerns a subject where a lot of different opinions are known to exist.

As pointed out by Mr. Achenbach, the paper presents a method which is basically sound, and has the potential of simplifying the estimation of the effect of heat storage in walls and other barriers.

Since the heating or cooling load consists of several components each of which is dependent on different types of basic variables, it is apparent that any method of simplifying heat load estimation is bound to have some limitations regarding coverage. The proposed method is meant for calculating that component of heat load which is due to transmission of heat through the walls and other barriers caused by the difference in the air temperatures. The authors agree with Messrs. Palmatier and Everetts that for cooling load in summer, differences in this component generally do not have a great effect on the total load calculations. The authors also agree with Mr. Signor that for heating load in winter this component plays an important part in the total load. This is due to the fact that the transmission load is generally a small percentage of the total cooling load in summer, whereas it is a fairly large percentage of the total heating load in winter. Consideration of periodic heat flow is, therefore, much more important in the estimation of winter load than the summer load, as also pointed out by Mr. Gilkey.

With reference to comments on solar heat load by Messrs. Brown, McGhan, Everetts and Gilkey, the proposed method is intended to be applicable both to heating and cooling load calculations and, therefore, does not include solar heat load. The solar heat load is more important for cooling load calculations than for heating load calculations and can be calculated by other methods. However, if it is considered desirable to combine the solar load with the transmission heat load, it can be done so and the factor  $F$  can be modified to combine both. Of course, it would then be necessary to have different factors  $F$  for summer and winter load calculations.

In reply to Professor Brown's comments, knowledge of time lag is required in some applications. The proposed method does not ignore the time lag. It gives the maximum load due to transmission. The time lag should be considered in arriving at the total load. Referring to the first example cited by Mr. Everetts, the authors suspect that the difference between 7.3 and 4.4 is most likely due to the  $U$  values being generally conservative.

The authors agree with Messrs. Heider and McGhan regarding need for wet bulb temperatures although this does not directly concern the proposed method.

Regarding Mr. McGhan's reference to the possible need for listing a large number of equivalent outside temperatures for a single structure, the authors feel that this would not be necessary. Two tables, one giving the values of  $F$  for common constructions and the other giving maximum and average daily temperatures, should suffice.

The authors agree with Mr. Palmatier regarding the importance of heat storage effect of materials within the building although the materials which are immediately within the outer skin and form a part of the barriers can be taken care of by the extension of the proposed method to non-homogeneous structures. The heat storage effect of furniture and other materials within the building cannot be accounted for by the proposed method, as those materials go through different temperature variations.

The total load consists of several components which have to be calculated separately. All one can hope to have is a simplified but accurate method for each component.



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## A THEORETICAL AND EXPERIMENTAL STUDY OF LIQUID-TO-LIQUID HEAT TRANSFER IN HOT WATER HEATERS

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THE INTENT of this research paper is to present experimental data on the quantitative evaluation of the heat transfer characteristics of water heaters in which transfer occurs from unagitated hot water to-and-through a tube wall to cooler water which is in forced circulation within the tube. In its broadest thermal characteristics the system is one which approaches classification as a liquid-to-liquid heat exchanger with natural convection controlling the film resistance on the outside of the tube and forced convection determining the film resistance on the inside; in most instances, as will be shown, the thermal resistance of the tube wall will be so small as to have no appreciable influence on the overall results (except insofar as the thickness of the wall alters the inside and outside areas of the transfer surface).

The paper reviews the semi-rational, dimensionless, empirical equations which seem applicable to the system and compares—by numerical example—the value of the overall coefficient of heat transfer obtained from application of the equations with that value obtained by computation from experimental data realized from the test set-up used for the experimental work. The test setup is described in detail and comparative test data are given for a single tube operating under controlled conditions with variable flow rates, and with varying liquid-to-liquid temperature differences.

### THEORETICAL RELATIONSHIPS

The system in question is simplified to involve transfer to a single tube of length sufficient to offset end effects, the tube being horizontal in a tank of hot water

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which is maintained at a practically fixed temperature without appreciable movement other than that due to thermal currents resulting from the transfer of heat; water within the tube undergoes temperature rise while traveling at a velocity sufficient to assure turbulent flow. Under these conditions heat transfer from the hot water in the tank to the cooler water within the tube would occur through a series path consisting of three thermal resistances: (1) outside film where the resistance would be controlled by the mechanism of free convection, (2) tube wall where the resistance would be controlled by the mechanism of conduction, (3) and inside film where the resistance would be controlled by the mechanism of forced convection.

The equation expressing rate of heat transfer from tank water to tube water would be

$$q = UA_o(t_t - t_w) = [A_o(t_t - t_w)] / [(A_o/h_i A_i) + (LA_o/kA_m) + (1/h_o)] \quad (1)$$

where

$q$  = rate of heat transfer, Btu per hour.

$t_t$  = temperature of the hot water within the tank, Fahrenheit degrees.

$t_w$  = mean temperature of the water within the tube, Fahrenheit degrees.

$A_o$  = outside surface area of the length of tube through which heat transfer occurs, square feet.

$A_i$  = inside surface area of the length of tube through which heat transfer occurs, square feet.

$A_m$  = mean area, in direction of heat transfer, of the length of tube through which heat transfer occurs, square feet.

$k$  = thermal conductivity of the tube wall, Btu per (hour) (square foot) (Fahrenheit degree per foot).

$L$  = thickness of the tube wall, feet.

$h_i$  = inside film coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree).

$h_o$  = outside film coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree).

$U$  = overall coefficient of heat transfer (expressed in terms of outside surface area), Btu per (hour) (square foot) (Fahrenheit degree).

$$= 1 / [(A_o/h_i A_i) + (A_o L/kA_m) + (1/h_o)] \quad (2)$$

For turbulent flow within the tube the recommended<sup>1</sup> relationship among the three controlling dimensionless parameters for determination of the inside film coefficient,  $h_i$ , is

$$(h_i D_i / k) = 0.023 (D_i V \rho / \mu)^{0.8} (c_p \mu / k)^{0.4} \quad (3)$$

The relationship recommended<sup>1</sup> for use in calculating the free convection value of the outside film coefficient,  $h_o$ , is

$$\frac{h_o D_o}{k_f} = 0.53 \left[ \left( \frac{D_o^3 \rho_f^2 \beta g \Delta t}{\mu_f^2} \right) \left( \frac{c_p \mu_f}{k_f} \right) \right]^{0.25} \quad (4)$$

<sup>1</sup> Heat Transmission, by W. H. McAdams (McGraw-Hill Book Co., Inc., 2nd ed., 1942).

In Equations 3 and 4 the terms not previously defined are:

$D_i$  and  $D_o$  = inside and outside tube diameters, feet.

$V$  = fluid velocity within the tube, feet per hour.

$\rho$  and  $\rho_f$  = fluid densities at, respectively, the mean temperature of the water within the tube and the mean temperature of the outside film.

$\beta$  = coefficient of volumetric expansion, cubic feet per cubic foot.

$g$  = acceleration due to gravity, feet per (hour).<sup>2</sup>

$\mu$  and  $\mu_f$  = absolute viscosity at, respectively, the mean temperature of the water within the tube and the temperature of the outside film, pounds per (foot) (hour).

$c_p$  = specific heat of water at the temperature in question.

$\Delta t$  = temperature difference across the outside film, Fahrenheit degrees.

*Test Conditions:* Equations 2, 3, and 4 permit determination of a theoretical value of the overall coefficient of heat transfer for a given system once the tube is specified with respect to size and material, the inside velocity is known, and the temperature relationships for the system have been specified. The physical properties of the fluid ( $\rho$ ,  $\mu$ ,  $k$ ,  $c_p$ ,  $\beta$ ) are all either functions of fluid temperature or are independent of temperature; hence, their values can be readily fixed when the mean fluid temperature is known.

For the particular test investigated here the test section consisted of a four-foot length of  $\frac{1}{2}$ -in. outside diameter copper tube with wall thickness of 0.028 in. and thermal conductivity of 220 Btu/(hr) (sq ft) (F deg/ft). The actual inside diameter was 0.444 in. and the outside diameter, 0.500 in. From tables for this standard tube it is noted that the capacity per foot of length is 0.008 gal. During the test the flow rate (obtained by a meter which had been calibrated with a weighing tank) was held constant at 3 gpm; the fluid velocity was therefore 3/0.008 or 375 fpm, which, in basic units, is 22,500 ft per hr.

After sufficient time had been allowed for the system to reach equilibrium, readings were taken at 2-min intervals over a 16-min test period. The temperatures of water entering and leaving the test section remained constant throughout this period at 20.70 C entering and 22.60 C leaving. The temperature of the water within the tank and surrounding the single tube varied approximately 0.1 deg with time and position, the average value being 57.73 C. Averaging and converting the Centigrade degrees experimental readings to Fahrenheit degrees, it follows that the mean temperature of water within the tube,  $t_w$ , was 71 F and the mean temperature of the water in the tank,  $t_t$ , was 135.9 F.

For a mean water temperature of 71 F the corresponding values of the physical properties are: density,  $\rho$ , of 62.2 lb/cu ft; viscosity,  $\mu$ , of 0.965 centipoises which equals 2.42 (0.965) or 2.335 lb/(ft) (hr); thermal conductivity,  $k$ , of 0.344 Btu/(hr) (sq ft) (deg/ft); specific heat,  $c_p$ , of 1.0 Btu/(lb) (deg).

*Theoretical Determination of the Overall Coefficient:* To determine the inside film coefficient for forced convection it is desirable to first evaluate the term of Equation 3 which corresponds to the Reynolds Number (since this term establishes the conditions of flow):

$$\text{Reynolds Number} = D_i V \rho / \mu = (0.444/12) (22500) (62.2)/2.335 = 22200$$

The next step is evaluation of the Prandtl Number:

$$\text{Prandtl Number} = c_p \mu / k = (1.0) (2.335) / 0.344 = 6.79$$

The Nusselt Number, appearing as the term of the left of Equation 3, is

$$\begin{aligned} \text{Nusselt Number} &= h_i D_i / k = (h_i) (0.444/12) / 0.344 = 0.023 (22200)^{0.8} (6.79)^{0.4} \\ &= 0.023 (3000) (2.15) = 148.4 \end{aligned}$$

The inside film coefficient,  $h_i$ , is then

$$h_i = (k/D_i) (148.4) = [(0.344)/(0.444/12)] (148.4) = 1380 \text{ Btu}/(\text{hr}) (\text{sq ft}) (\text{F deg})$$

Calculations of the outside film coefficient require evaluation of the Grashof Number (the first bracketed term on the right side of Equation 4) and of the Prandtl Number, both dimensionless parameters to be evaluated with physical properties at the mean temperature of the outside film. But the thermal resistance for free convection through the outside film would be expected to greatly exceed the series resistances for forced convection through the inside film and for conduction through the tube wall. Hence, the greater part of the water-to-water temperature drop will occur in the outside film and the mean film temperature can therefore be taken, for a first approximation, as halfway between the temperature of water in the tank and the mean water temperature in the tube; the mean film temperature,  $t_f$ , is thus equal to  $(135.9 + 71)/2 = 103.5 \text{ F}$ . Values of the fluid properties at this temperature are: density,  $\rho_f$ , of 61.9 lb/cu ft; viscosity,  $\mu_f$ , of 0.648 centipoises or 1.568 lb/(ft) (hr); thermal conductivity,  $k_f$ , of 0.360 Btu/(hr) (sq ft) (F deg/ft). The value of the coefficient of volumetric expansion  $\rho$ , is  $0.115(10)^{-3}$  and of the acceleration due to gravity,  $g$ , is  $4.17(10)^8 \text{ ft}/(\text{hr})^2$ . The mean temperature difference between the heating and cooling fluids is  $135.9 - 71 = 64.9 \text{ F}$  and this is approximately (refer to following section, *Corrected Theoretical Coefficient*) equal to the temperature difference,  $\Delta t$ , across the outside film.

Then evaluating the Grashof Number,

$$\begin{aligned} \text{Grashof Number} &= (0.500/12)^3 (61.9)^2 (0.115) (10)^{-3} (4.17) (10)^8 (64.9)/(1.568)^2 \\ &= 351,500 = 0.351 (10)^6 \end{aligned}$$

The Prandtl Number, for outside film conditions, is

$$\text{Prandtl Number} = c_{pft} / k_f = (1.0) (1.568) / (0.360) = 4.36$$

Then substituting into Equation 4 to obtain the Nusselt Number applicable to the outside film,

$$\text{Nusselt Number} = 0.53 [(351,500) (4.36)]^{0.25} = 0.53 (35.15) = 18.65$$

The resultant value of the outside film coefficient is

$$h_o = (k_f/D_o) (18.65) = 18.65 (0.360)/(0.500/12) = 161.2 \text{ Btu}/(\text{hr}) (\text{sq ft}) (\text{F deg})$$

The overall coefficient can now be calculated by Equation 2, noting that ratios of tube diameters in inches are equal to ratios of surface areas in square feet, (and taking the mean tube diameter as the arithmetical average of inside and outside diameters).

$$U = \frac{1}{\frac{0.500}{(0.444)(1380)} + \frac{(0.500)(0.28/12)}{(0.472)(220)} + \frac{1}{161.2}} = 1/(0.000817 + 0.0000112 + 0.00620) \\ = 1/0.007028 = 142.2 \text{ Btu/(hr) (sq ft) (F deg)}$$

Note that the reciprocal of the overall coefficient is the total resistance and the three individual additive terms which make up the total resistance are the resistances of, respectively, the inside film, the tube wall, and the outside film; all resistances are evaluated, as is the overall coefficient, in terms of unit area of the outside surface of the tube.

*Corrected Theoretical Coefficient:* The given theoretical evaluation of the overall coefficient was based on the assumption that the entire water-to-water temperature drop occurred through the outside film. But the actual temperature drop through the outside film is to the water-to-water temperature drop as the outside film resistance is to the total thermal resistance. The resistances were determined in the above section; hence a more accurate approximation to the drop through the outside film is obtainable:

$$\text{Film temperature drop} = (64.9) (0.00620)/(0.007028) = 57.2 \text{ F}$$

Noting that the one-quarter power of the film temperature drop enters into the evaluation of  $h_o$ , a corrected value of the outside film coefficient is

$$h_o = 161.2 (57.2/64.9)^{0.25} = 156.4$$

The resistance of the outside film then becomes  $1/156.4 = 0.00639$  and the corrected value of the overall coefficient is

$$U = 1/(0.000817 + 0.0000112 + 0.00635) = 1/0.007218 \\ = 138.5 \text{ Btu/(hr) (sq ft) (F deg).}$$

\* Based on the new value of the outside film resistance a third approximation gives:

$$\text{Film temperature drop} = (64.9) (0.00639)/(0.007218) = 57.5$$

and the corrected outside film coefficient is

$$h_o = 161.2 (57.5/64.9)^{0.25} = 156.5$$

This value differs inappreciably from the  $h_o$  determined from the previous approximation, 156.4; hence the overall coefficient obtained from the second approximation, 138.5 Btu per (hr) (sq ft) (F deg), can be accepted as the most accurate value obtainable from the given equation.

*Experimental Determination of the Overall Coefficient:* It has already been noted that under test conditions the water flowing through the 4-ft length of tube underwent a temperature rise,  $(\Delta t)_h$ , from 20.7 C to 22.6 C, or gained 3.42 F deg. Flow rate was 3 gpm, so the total energy transfer amounted to:

$$Q = Wc (\Delta t)_h = (3) (8.345) (1) (3.42) = 85.6 \text{ Btu/min} \dots (5)$$

But the test section having 0.500-in. diameter has 0.131 sq ft of outside surface per foot length; the rate of heat transfer is therefore:

$$q = 60 (Q)/(4) (0.131) = (60) (85.6)/(4) (0.131) = 9800 \text{ Btu/(hr) (sq ft)}$$

Then substituting into Equation 1,

$$q = UA_o (t_i - t_w) = 9800 = U (1) (135.9 - 71) = 64.9U$$

giving

$$U = 151.0 \text{ Btu/(hr) (sq ft) (F deg).}$$

*Comparison of Theoretical and Experimental Results:* The calculated value, 138.5 Btu/(hr) (sq ft) (F deg), of the overall coefficient of heat transfer differs from the experimental value, 151.0 Btu/(hr) (sq ft) (F deg), by slightly more than 8 percent. This result shows that the equations for calculating the inside

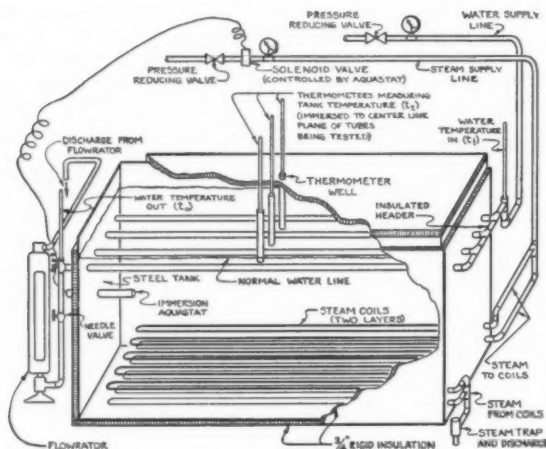


FIG. 1. ARRANGEMENT FOR TESTING WATER-TO-WATER HEAT TRANSFER UNDER CONDITIONS OF FREE CONVECTION OUTSIDE THE TUBES

and outside film coefficients can be expected to give results of the correct order of magnitude. This fact demonstrates the possibility of accurately extrapolating limited experimental results for water heaters over a range of diameters, inside fluid velocities, and mean fluid temperatures.

#### EXPERIMENTAL PROCEDURE

Fig. 1 illustrates diagrammatically the equipment arrangement used for the experimental work. The insulated chamber permitted use of a 48-in. straight length of each of the four test tubes which are located near the top of the chamber. The tubes could be tested as a group or, by means of control valves (not shown) on the discharge side, could be tested individually. Water supply to the units under test was from the city water system, the flow rate being accurately maintained at a fixed value by fixed setting of the inlet pressure reducing valve and

controlled setting of the needle valve on the discharge side. Flow rate was measured by means of a weighing tank-calibrated flow meter as shown. The water within the tank was maintained at a fixed temperature during each test. Heat to the tank was supplied by two sinuous coils located near the bottom of the tank, one above the other; each coil has ten rows of  $\frac{7}{8}$ -in. outside diameter copper tube,  $2\frac{1}{2}$ -in. center-to-center,  $45\frac{1}{2}$ -in. long between return bends; the lower coil is  $1\frac{1}{2}$ -in. above the bottom of the tank and the upper coil,  $1\frac{1}{2}$ -in. above the lower.

As heat flows from the hot tank water to the cooler water within the test tubes it is evident that circulation of the water within the tank will occur by free con-

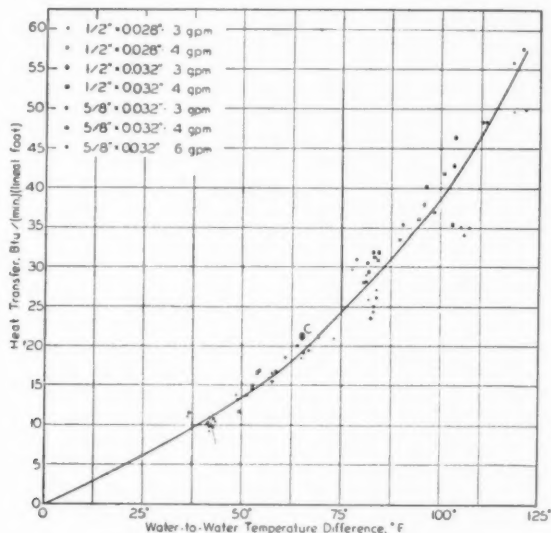


FIG. 2. RELATION BETWEEN WATER TEMPERATURE DIFFERENCE AND HEAT TRANSFER RATE RESULTING FROM EXPERIMENTAL STUDY

vection as a result of the vertical variation in density. One major purpose of the study was to determine whether or not the degree of circulation was sufficient to invalidate application of the basic (Equation 4) relationship for free convection. No agitation was provided for the tank water, but by observation the degree of circulation would be expected to increase as some function of the rate of heat transfer.

#### EXPERIMENTAL RESULTS

Fig. 2 summarizes the experimental data obtained from some 130 tests of two diameters (and two wall thicknesses) of copper\* tube. Each test was run at fixed

\* Although all tests were run using copper tube, the results can be readily applied to ferrous pipe or to alloy tubes by correcting the tube wall resistance; the procedure for separating the three thermal resistances of the system is demonstrated in the example given in the section, Theoretical Relationships.

tank temperature and at fixed flow rate; variation in tank temperature among the tests was over the range from 86 F to 186 F; variation in flow rate among the tests was from 1 to 6.4 gpm.

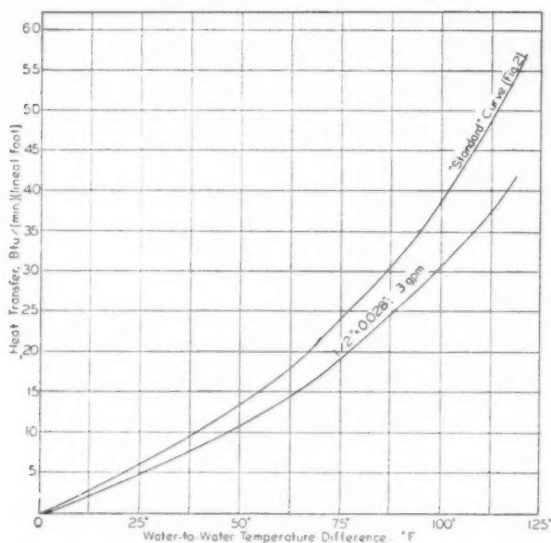


FIG. 3. STANDARD CURVE FROM EXPERIMENTAL DATA OF FIG. 2 RELATIVE TO THE CURVE FOR  $\frac{1}{2}$ -IN OD TUBE AT 3 GPM

In seeking a correlation of the experimental data it would be expected (by observation of the terms in Equations 1, 2, 3, and 4) that three variables would require consideration.

1. Temperature of water within the tube and within the tank and mean temperature difference from water-to-water.
2. Diameter and wall thickness of tube.
3. Velocity of water within the tube.

But consideration of the example given in the section, Theoretical Relationships, will show that the thermal resistance due to the outside free convection film accounts for  $(100 \times 0.00639/0.007218)$  73 percent of the total thermal resistance of the system. For tube velocities higher than that of the example (3 gpm) the percentage would exceed 73 percent; whereas for lower velocities the inside film would have increased significance. Thus the rate of heat transfer would be expected to be largely controlled by the factors which determine the outside film coefficient.

Referring to Equation 4, note that the diameter—a fraction—enters in the third power, hence will have small influence on  $h_o$ . Over the usual temperature range the variation in one quarter power of the various physical characteristics will not be great. Thus it follows that the major variable which would be expected to influence the outside film is the water-to-water temperature difference. Then, for velocities between 3 and 6 gpm, a rough correlation might be expected between rate of heat transfer and water-to-water temperature difference, neglecting actual water temperatures, tube diameter, tube wall thickness, and velocity of water within the tube; Fig. 2 presents such a correlation.

In interpreting Fig. 2 it should be remembered that the experimental points would not be expected to fall on a single curve; hence departures are frequently due to differences in velocity, temperature, or tube type. The point identified by the letter C is the one which was checked by the theoretical method of the preceding section; where accuracy in design requires, other points can be readily checked by the same method. The significance of the curve indicated in Fig. 2 resides in its usefulness as a means of obtaining a rapid and simple approximation to heat transfer rates in water-to-water heaters with external free convection.

Fig. 3 presents the curve of Fig. 2 together with a similar curve for  $\frac{1}{2}$ -in. outside diameter (0.028-in. wall thickness) tube with inside velocity of 3 gpm.

### CONCLUSION

Based on an experimental investigation of water-to-water heat transfer it is found that the empirical equations commonly used for turbulent flow within a tube and for free convection outside the tube permit evaluation of film coefficients, hence of the overall coefficients of heat transfer, which agree—within the limits of experimental error—with the values obtained by test under conditions simulating those existing within a water heater. Further, the demonstrated major influence of the outside film thermal resistance is such that a reasonably accurate approximate correlation can be obtained between water-to-water mean temperature difference and the rate of heat transfer. The approximate correlation—accurate to within better than 10 percent for the greater part of its range—is valid over a temperature range from 86 F to 186 F, a velocity range from 3 to 6 gpm (in  $\frac{1}{2}$ -in. or  $\frac{5}{8}$ -in. outside diameter copper tubes) and is applicable to various tube wall thicknesses.

Where accuracy greater than that of the correlation curve is required, the paper shows that the theoretical relationships can be applied; similarly, for ferrous pipe or alloy tubes the theoretical relationships permit correction of the experimental data to obtain design values of the rate of heat transfer.

### DISCUSSION

M. L. GHAI, West Hartford, Conn.: I will appreciate the author's comments regarding the range of hydraulic radii over which the data presented could be considered applicable for practical purposes within an accuracy of about 10 percent. Have the authors experimented with heaters having narrow water passages with hydraulic radii of the order of  $\frac{1}{32}$  in.?

AUTHOR'S CLOSURE (Professor Hutchinson): No, we haven't in connection with the present research done anything of the type which the question raised.

I might add in passing that such work is being done by the heat transfer group at the university in terms of a more formal presentation, but in this paper we deliberately stayed away from what might be called the more technical aspects in the sense of variations of geometric angle and things of that kind because we simply wanted to find out from the overall statement how we could explain the relatively theoretical equations to a very simple problem.

I mention that not by way of apology but by way of explaining that we deliberately avoided all of the special conditions which applied to such systems because we felt it would be nice to find out how the relatively complex equations apply to a simple barnyard problem. So that was the intent of the paper and that is the basis.



**1533**

## SOLAR RADIATION DURING CLOUDLESS DAYS

This paper is the result of researches sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., The Minnesota Institute of Research and the Graduate School, of the University of Minnesota, conducted through the Department of Mechanical Engineering of the University of Minnesota.

By J. L. THRELKELD\* AND R. C. JORDAN\*\*, MINNEAPOLIS, MINN.

THE WORLD'S present energy requirements for food, power and heat approximate 60 billion Btu per day; solar radiation intercepted by the earth each day equals about 5600 million million Btu. It is this vast differential which, in view of the rapidly dwindling fossil fuel deposits of the earth, is accelerating exploitation of solar energy. Of the sun's rays reaching the earth's outer atmosphere, only 16 percent reaches the land areas of the world, but the one percent intercepted by continental United States is equivalent to about 800 tons of coal per acre per year.

Although utilization of solar energy poses many technical problems, the stakes are large. Currently, about 40 percent of all energy consumed in the United States is involved in space heating. Since space heating is accomplished at comparatively low energy levels, the technical problems are not as great as in other areas. However, space heating includes only a fraction of the potential uses to which solar energy may be placed eventually. Some, as with the potential use of solar energy for space heating, involve replacement of fuels by solar energy. In this category are also included the generation of electricity by thermopiles consisting of thermocouples with high thermoelectric effect and the generation of electricity by photogalvanic effects, or by so-called *solar batteries* such as the one under development by Bell Telephone Laboratories which utilizes a silicon *p-n* junction, similar to a junction transistor. Upon the absorption of light on the *battery* surfaces a potential is developed between the ends of the crystals through the liberation and migration of free-to-move negative and positive electric charges.

There are many potential additional uses, however, which may, upon development, extend our standard of living through new energy applications. For example, in order to reduce agricultural wastes, the final drying of such crops as corn and hay is, in some instances, now being accomplished artificially through oil-

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fired drying systems. Since the intermittency of solar energy is here no disadvantage, it is quite possible that solar drying systems could extend this practice at a marked reduction in processing cost. Another potential application is the drying of peat for fuel. Dried peat is used as a power fuel in Russia, Germany, Ireland and, to a lesser extent, in Denmark, Sweden and Finland. In this country it is not yet economically competitive with other fuels, but the fact remains that about 14 billion tons of peat are potentially available. If this can be dried economically, our fossil fuel reserves will be extended appreciably. A third example of the possible extension of living standards concerns the current pilot plant attempts to culture the single celled alga, *Chlorella*, by circulation in a closed nutrient solution periodically exposed to full sunlight. Since algae are capable of absorbing intense light during brief periods and later utilizing this energy in photosynthesis, this may develop more efficient sources of protein than any of our current farming methods. Potentially, *Chlorella* culture can provide a protein yield per acre 200 times greater than soybeans.

In the adaptation of solar energy to the solution of engineering problems, three factors are involved: (1) determination of the availability and reliability of the energy supply, (2) determination of the energy requirements of the problem to be solved and (3) development of efficient collection or conversion equipment.

Currently, one of the most important problems involves an evaluation of the energy source, and this paper provides an extension of the present knowledge. Engineering applications will require information as to the incidence of solar radiation outside the atmosphere, at various depths of the atmosphere, at the earth's surface during cloudless days, and at the earth's surface during all days. Test results reported by Maria Telkes and Eleanor Raymond<sup>1</sup> on the solar house at Dover, Mass. show that almost 94 percent of the energy collected in the month of February 1949 occurred during 15 days of above average incident radiation. Thus, it is possible that the extent and reliability of cloudless day radiation may prove to be of major importance in the design of equipment for solar energy collection and conversion. This paper provides an engineering estimate of the incidence of solar radiation at the earth's surface during cloudless days.

#### METHODS OF CALCULATION

This section discusses methods of calculating solar and sky radiation during cloudless days at the latitudes and for the conditions applicable to the United States.

*Direct Solar Radiation Perpendicular to Sun's Rays:* In a very important paper, Parry Moon<sup>2</sup> has presented a method of calculating the direct solar irradiation of a surface normal to the sun's rays during cloudless days for any elevation and for varying amounts of water vapor and dust in the atmosphere. Moon has calculated spectral irradiation curves for assumed atmospheric conditions of sea level, 20 mm of precipitable water, 300 dust particles per cc, and 2.8 mm of ozone. The ASHAE has taken these data as a standard for clear atmospheres and typical summer conditions and has correlated them in *THE GUIDE*<sup>3</sup> showing direct solar radiation at normal incidence as a function of the solar altitude angle.

F. W. Hutchinson and W. P. Chapman<sup>4</sup> have applied Parry Moon's calculation method to an assumed standard winter atmosphere of sea level, 33 F dew-point temperature, 300 dust particles per cc and 2.8 mm of ozone. Their results are

<sup>1</sup> Exponent numerals refer to References.

correlated by a curve showing direct solar radiation at normal incidence as a function of the solar altitude angle.

*Sky Radiation:* Parmelee<sup>5</sup> has contributed important information on the irradiation of horizontal and vertical surfaces by diffuse or sky radiation. However, many of the calculations for this paper were made prior to publication of Parmelee's paper and sky radiation values were taken from curves presented in a previous publication<sup>6</sup> by the authors. Since during cloudless days the magnitude of diffuse radiation is generally small compared to the direct solar component, this circumstance has little effect on the results presented in this paper.

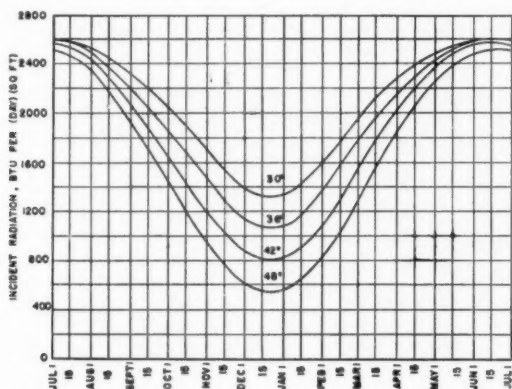


FIG. 1. DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLOUDLESS DAYS AT VARIOUS NORTH LATITUDES

*Solar and Sky Radiation—Horizontal Surfaces:* The intensity of direct radiation incident upon a horizontal surface is given by the relation

$$I_H = I_N \sin \beta \quad (1)$$

where

$I_H$  = direct solar radiation incident upon horizontal surface, Btu per (hour) (square foot).

$I_N$  = direct solar radiation incident upon a surface normal to sun's rays, Btu per (hour) (square foot).

$\beta$  = solar altitude angle.

The total hourly rate of radiation is found by adding the diffuse radiation to that calculated by Equation 1. The total radiation incident upon the surface during the day is found by summing up the hourly rates using Simpson's rule.

Fig. 1 shows daily total solar and sky radiation incident upon a horizontal surface during cloudless days for north latitudes of 30, 36, 42, and 48 deg. In calculating the curves of Fig. 1, computations for May through September were made using Moon's values<sup>3</sup> for  $I_N$  in Equation 1 and Parmelee's values<sup>5</sup> for diffuse radia-

tion, while for October through April Hutchinson's and Chapman's values<sup>4</sup> were used for  $I_N$  and the authors' values were used for diffuse radiation. The two sets of daily totals were plotted with the spring and autumn values of each set adjusted to give the smooth curves of Fig. 1.

Fig. 1 shows that maximum energy incidence on a horizontal surface occurs at 30 deg north latitude with a year peak of about 2600 Btu per (day) (sq ft) during the June and July period. There is relatively little difference in radiation incidence during the summer months between the several latitudes. Thus, the irradiation of a horizontal surface during cloudless days is fairly uniform in the United States except in mountainous locations and in industrial regions where the atmosphere is heavily polluted with dust. Because of much smaller solar altitude angles in winter, energy incidence is much lower in winter months with

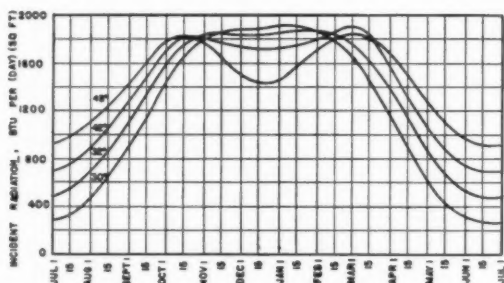


FIG. 2. DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE DURING CLOUDLESS DAYS AT VARIOUS NORTH LATITUDES

a minimum of about 550 Btu per (day) (sq ft) occurring in December at 48 deg north latitude.

*Solar and Sky Radiation—South-facing Vertical Surfaces:* The intensity of direct radiation incident upon a south-facing vertical surface is given by the relation

$$I_{BV} = I_N \cos \beta \cos \alpha \dots \dots \dots (2)$$

where

$I_{BV}$  = direct solar radiation incident upon south-facing vertical surface, Btu per (hour) (square foot).

$I_N$  = direct solar radiation incident upon a surface normal to sun's rays, Btu per (hour) (square foot).

$\beta$  = solar altitude angle.

$\alpha$  = wall solar azimuth angle.

The total hourly rate of radiation is found by adding the diffuse radiation to the direct component calculated by Equation 2 with daily totals of incidence found by summing up the hourly rates using Simpson's rule.

Fig. 2 shows daily total solar and sky radiation incident upon a south-facing vertical surface during cloudless days for north latitudes of 30, 36, 42 and 48 deg.

The same sources of information for normal incidence radiation and diffuse radiation were used in calculating Fig. 2 as for Fig. 1. Fig. 2 shows that maximum winter incidence and minimum summer incidence upon a south-facing vertical surface occurs at 30 deg north latitude. A peak of about 1900 Btu per (day) (sq ft) occurs on about January 1 at 30 deg while peaks of about 1800 Btu per (day) (sq ft) occur on October 15 and March 1 at 48 deg north latitude. Incidence during the winter months is fairly uniform except at 48 deg latitude where a minimum occurs on December 22.

Fig. 3 shows the ratio of south-facing vertical to horizontal radiation during cloudless days and was calculated from the data for Figs. 1 and 2. Fig. 3 illustrates the superiority of the south-facing surface for winter collection of sunshine, especially at the more northerly latitudes.

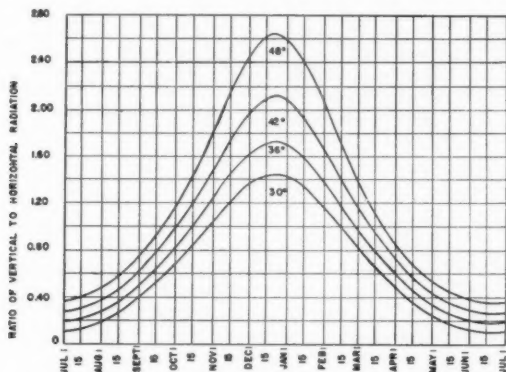


FIG. 3. RATIO OF DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE TO THAT INCIDENT UPON A HORIZONTAL SURFACE DURING CLOUDLESS DAYS AT VARIOUS NORTH LATITUDES

*Solar and Sky Radiation—South-facing Tilted Surfaces:* If a south-facing surface is tilted from the vertical position by an angle  $\phi$ , the intensity of direct solar radiation incident upon this surface is given by the relation

$$I_{ST} = I_N (\sin \beta \sin \phi + \cos \beta \cos \alpha \cos \phi) \quad (3)$$

where

$I_{ST}$  = direct solar radiation incident upon south-facing tilted surface, Btu per (hour) (square foot).

$I_N$  = direct solar radiation incident upon a surface normal to sun's rays, Btu per (hour) (square foot).

$\beta$  = solar altitude angle.

$\alpha$  = wall solar azimuth angle.

$\phi$  = angle of tilt from vertical position.

Combining Equations 1, 2 and 3 gives

$$I_{ST} = I_H \sin \phi + I_{SV} \cos \phi \quad (4)$$

An optimum tilt angle may be found by differentiating Equations 3 or 4 with respect to  $\phi$ , and equating to zero, considering  $I_N$ ,  $\beta$  and  $\alpha$  as constants. The result is

$$\tan \phi = \tan \beta / \cos \alpha = I_H / I_{av} \quad (5)$$

Since  $\beta$  and  $\alpha$  are variables, the optimum tilt angle also varies. In systems utilizing solar energy for space heating, practical considerations dictate a fixed position for the collector. Thus, an optimum tilt must be determined over a period of time rather than for a particular instant. Since the December-February period

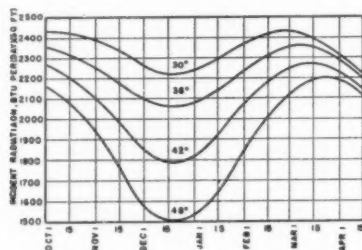


FIG. 4. DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON A SOUTH-FACING SURFACE TILTED FROM THE HORIZONTAL AT AN ANGLE EQUAL TO THE LATITUDE PLUS 21 DEG DURING CLOUDLESS DAYS AT VARIOUS NORTH LATITUDES

is critical for a solar heating system, the method was adopted to assume the tilted surface perpendicular to the sun's rays at 12:00 noon on January 15. This requires that the tilt angle  $\phi$  be equal to the solar altitude angle for this time. The tilt angles thus determined are shown in Table 1. It is observed that in every case the tilt angle from the horizontal position is closely equal to the latitude plus 21 deg.

TABLE 1—TILT ANGLES REQUIRED TO MAKE A SOUTH-FACING SURFACE NORMAL TO SUN'S RAYS AT NOON ON JANUARY 15

NORTH LATITUDE DEG	TILT ANGLE FROM VERTICAL		TILT ANGLE FROM HORIZONTAL	
	DEG	MIN	DEG	MIN
30	38	51	51	9
36	32	49	57	11
42	26	49	63	11
48	20	49	69	11

Fig. 4 shows daily total solar and sky radiation incident upon a south-facing surface tilted by the angles shown in Table 1 for the winter period October 1 to

April 15. In the construction of Fig. 4, hourly rates of direct solar radiation were calculated by Equation 4. Since there is no information on diffuse radiation incident upon tilted surfaces, this component was estimated by the linear relation

$$I_{BT,d} = I_{SV,d} + (I_{H,d} - I_{SV,d}) \phi/90 \quad . . . . . (6)$$

where

$I_{ST,d}$  = diffuse radiation incident upon tilted surface, Btu per (hour) (square foot).

$I_{SV,d}$  = diffuse radiation incident upon south-facing vertical surface, Btu per (hour) (square foot).

$I_{H,d}$  = diffuse radiation incident upon horizontal surface, Btu per (hour) (square foot).

$\phi$  = tilt angle from vertical position, degrees.

The total hourly rates found by adding Equations 4 and 6 were summed by Simpson's rule to give daily totals.

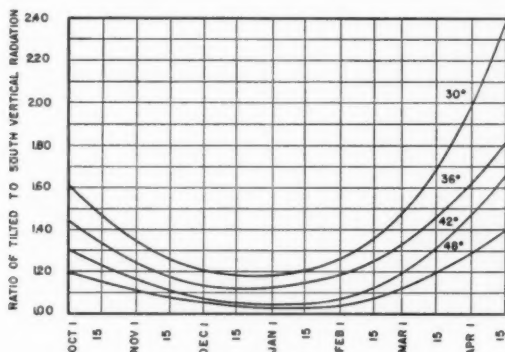


FIG. 5. RATIO OF DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON A SOUTH-FACING SURFACE TILTED FROM THE HORIZONTAL AT AN ANGLE EQUAL TO THE LATITUDE PLUS 21 DEG TO THAT INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE DURING CLOUDLESS DAYS AT VARIOUS NORTH LATITUDES

Fig. 4 shows that the incidence of radiation upon a south-facing surface tilted from the horizontal by an angle equal to the latitude plus 21 deg is a maximum in the southern part of the United States with the effect of latitude being most significant in the month of December.

Fig. 5 shows the ratio of south-facing tilted to south-facing vertical radiation during cloudless days for the October 1 to April 15 period and was calculated from

the data for Figs. 2 and 4. Fig. 5 shows that the advantage of tilting is most pronounced in the more southerly latitudes where an increase of about 20 percent occurs during the December-January period at 30 deg north latitude. Fig. 5 also shows that in the more northerly latitudes during the December-January period very little increase of incident radiation results from tilting the south-facing surface.

#### COMPARISON OF CALCULATED AND RECORDED RADIATION DURING CLEAR DAYS

The calculated results previously presented are for an assumed standard atmosphere at sea level. The dust content of 300 particles per cc is representative of a

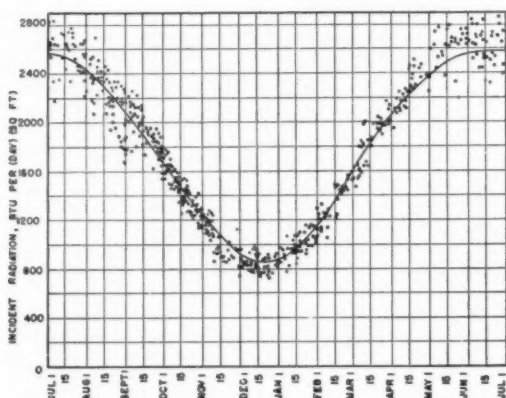


FIG. 6. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT LINCOLN, NEB.

reasonably clean atmosphere. As a check on the calculations, comparisons were made with recorded radiation published by the U. S. Weather Bureau for Lincoln, Neb.; Madison, Wis.; Columbia, Mo.; Rapid City, S. D.; Nashville, Tenn.; and Blue Hill, Mass. The plotted points shown in Figs. 6-12 are daily totals of recorded radiation for days with 0-2 cloudiness and 95-100 percent sunshine.

Fig. 6 shows a comparison of the curve of calculated solar and sky radiation incident upon a horizontal surface for the latitude of Lincoln, Neb. (40 deg, 49 min) with plotted points of clear day radiation as recorded on a horizontal surface at Lincoln for the 10-year period 1944-1953. It is evident that the assumed standard atmosphere calculation gives a very good average correlation.

Fig. 7 shows a similar comparison of calculated and recorded radiation incident upon a horizontal surface for Madison, Wis., latitude 43 deg, 8 min, for the 10-year period 1944-1953. It is seen that the calculated curve gives a good average correlation except in the May-August period when it is about 5 percent too low.

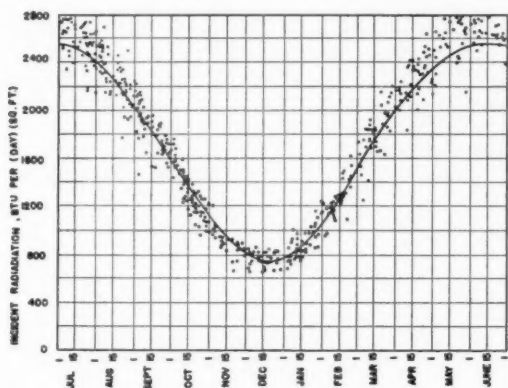


FIG. 7. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT MADISON, WIS.

Fig. 8 shows a comparison of calculated and recorded radiation incident upon a horizontal surface for Columbia, Mo., latitude 38 deg, 57 min, for the 10-year period 1944-1953. The calculated curve results in a reasonably good correlation for the winter months but is consistently low by 5 to 10 percent during the April-September period.

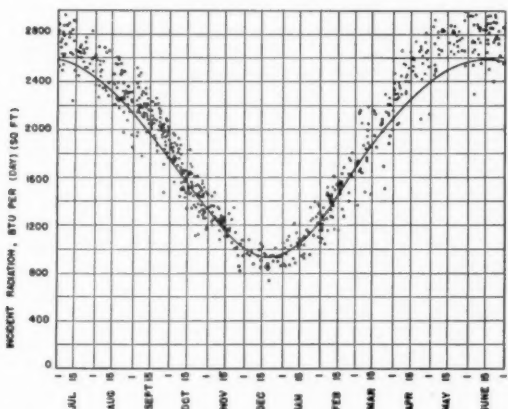


FIG. 8. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT COLUMBIA, Mo.

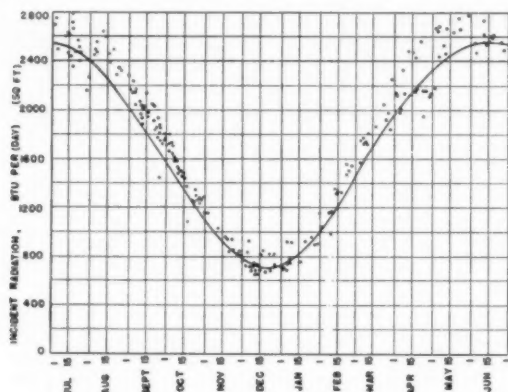


FIG. 9. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT RAPID CITY, S. D.

Fig. 9 shows a comparison of calculated and recorded radiation incident upon a horizontal surface for Rapid City, S. D., latitude 44 deg, 9 min, for the five-year period 1949-1953. Although there are insufficient points for a representative comparison, it appears that the calculated curve is consistently low by about 5 to 10 percent.

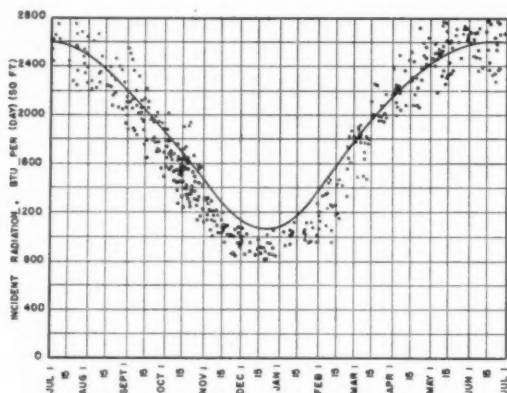


FIG. 10. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT NASHVILLE, TENN.

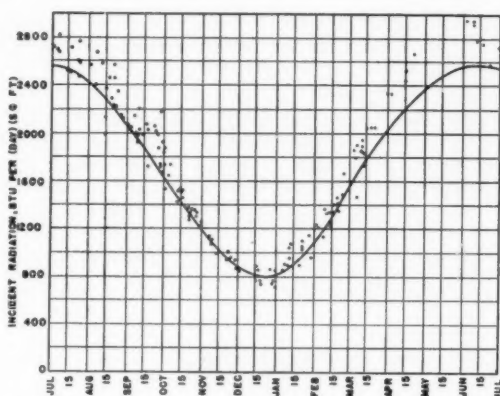


FIG. 11. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING CLEAR DAYS AT BLUE HILL, MASS.

Fig. 10 shows a comparison of calculated and recorded radiation incident upon a horizontal surface for Nashville, Tenn., latitude 36 deg, 9 min, for the eight-year period 1944-1948, and 1951-1953. It is seen that the calculated curve gives a good average correlation for the April-September period but is high by some 15 to 20 percent during the winter months. According to Hand<sup>7</sup>, considerable smoke contamination in winter months occurs at Nashville and this may account for the higher radiation as indicated by the calculated curve.

Fig. 11 shows a comparison of calculated and recorded radiation incident upon

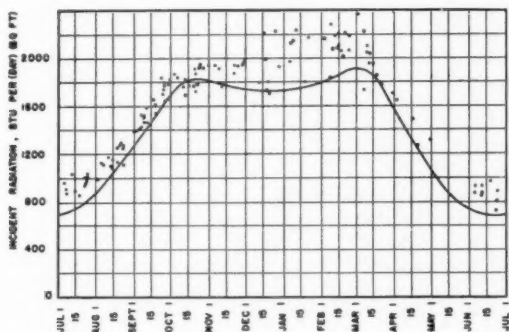


FIG. 12. COMPARISON OF CALCULATED AND RECORDED SOLAR AND SKY RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE DURING CLEAR DAYS AT BLUE HILL, MASS.

a horizontal surface during clear days for Blue Hill, Mass. The latitude of Blue Hill is 42 deg, 13 min. The plotted points represent but five years of data (1949-1953) which are inadequate for a representative comparison. The calculated curve compares favorably with the recorded data during winter months but is consistently low during the March-September period by some 5-10 percent.

Fig. 12 shows a comparison of calculated and recorded radiation incident upon a south-facing vertical surface during clear days for Blue Hill, Mass. The plotted points are for the same years of data as Fig. 11 and thus are not sufficient for an average comparison. During the April-October period the calculated curve is low by some 5 to 10 percent or by a similar deficiency to the calculated curve of Fig. 11 for a horizontal surface. However, in the December-February period the calculated curve is much too low, indicating on some days a considerable increase by reflection from the ground.

### CONCLUSIONS

The authors believe that the calculated curves on cloudless day radiation shown in Figs. 1 through 5 are sufficiently accurate for engineering purposes for all localities in the United States except mountainous locations and highly industrialized regions. Comparison of calculated and recorded radiation incident upon a horizontal surface for six localities indicate average correlation by the calculation method within about 5 to 10 percent except at Nashville, Tenn. The calculated curves presented in this paper are based upon the basic equations developed by Parry Moon<sup>2</sup> and these equations may be applied to any type of atmosphere including mountainous and industrial regions.

Maximum incidence of solar and sky radiation on a horizontal surface during cloudless days occurs in the southern part of the United States, although in summer months the radiation is fairly uniform throughout the country. In winter months a significantly greater incidence occurs in the more southerly latitudes.

Maximum incidence of solar and sky radiation on a south-facing vertical surface during cloudless days occurs in the southern part of the United States in winter and in the northern part of the country in summer. The ratio of south-facing vertical to horizontal surface radiation incidence is larger throughout the year at the more northerly latitudes, reaching a maximum in December.

The tilting of a south-facing surface in winter months may increase the incidence of radiation during cloudless days by 20 percent or more over that of a south-facing vertical surface in the more southerly latitudes. In the northerly latitudes, the increase is much smaller, being less than 5 percent for December and January at 48 deg north latitude.

Further pyrheliometric studies are necessary to establish the effect of ground reflection of radiation on vertically-placed and tilted surfaces. In winter months a considerable increase of incident radiation beyond the calculated amount may occur by reflection from a snow-covered ground surface. The advantages of tilting a south-facing surface are dependent also upon ground reflection, since a tilted surface receives less reflected radiation than a vertically-placed surface.

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## DISCUSSION

C. F. KAYAN, New York, N. Y.: This very interesting paper brings out in the early part a statement that arouses a bit of curiosity. I wonder if we can have an interpretation on it.

This statement: "Currently about 40 percent of all energy consumed in the United States is involved in space heating."

Bearing in mind that we do use a lot of energy for industrial processes, also for transportation—the automobile, by way of example; that the United States is heated in the wintertime only in certain sections—we have a large spread of territory in the northern part and the southern part—40 percent seems an unusually large percent of the total energy consumed. I am rather consumed with curiosity for an interpretation of it.

ROGER HAINES, Albuquerque, N. M.: We in the Southwest are vitally interested in the subject matter of this paper and similar papers that preceded it, because we think that in our locality, New Mexico and Arizona, the climate is ideally suited for applications of solar energy to space heating, and we hope in a couple of years that we might have something for you on this line.

I was talking to Dr. Threlkeld a moment ago and he suggested that you might be interested in some of the direct results of solar radiation in a climate such as ours.

We have recently finished a 12-story office building in Albuquerque which is heated by means of a heat pump. The south and north sides of the building are about 70 percent glass and our solar altitude in December is about 30 deg, so that the incident angle is fairly great on the south facing surface.

One morning recently with a temperature of 18 F. at nine o'clock in the morning, when one might think that the flywheel effect would call for heating, cooling was required on the entire south side of the building, even in the unoccupied areas.

M. L. GHAI, West Hartford, Conn.: The authors have presented some of the basic information needed for investigating the feasibility of utilizing solar energy. I certainly hope that more of this type of work can be done so we can determine more precisely what we can do with this virtually inexhaustible source of energy.

The curves presented in this paper also indicate that in the middle of January the incident solar energy is higher than the energy received in December. As far as space heating with solar energy is concerned, January is important since the degree-days of heating are generally maximum about the middle of January.

AUTHOR'S CLOSURE (Doctor Threlkeld): We appreciate very much the comments of Professor Kayan, Mr. Haines and Dr. Ghai.

I am very interested in Mr. Haines' comments. It is encouraging to hear that a commercial application of solar energy for space heating is now being designed. I

agree that Arizona and New Mexico are excellent locations for solar utilization systems because of their high percent sunshine and fairly mild winter temperatures.

In regard to Dr. Ghai's comment, which is very pertinent, I may say that the incidence on a horizontal surface is at a minimum on about December 22 for two reasons. First, at this date the sun's altitude is lowest of the year and consequently the length of path of the sun's rays through the atmosphere is greatest. Thus, the depletion effect of the atmosphere is at a maximum. Secondly, as shown by Equation 1 of our paper, the direct solar radiation is calculated as  $I_n \sin \beta$  and on December 22, the values of  $\sin \beta$  are at a minimum.

Fig. 2 shows that the incidence on a south-facing vertical surface is fairly uniform during December and January except at the more northerly latitudes. As shown by Equation 2, the calculation of direct solar radiation incident on a south-facing vertical surface involves the term  $\cos \beta$ . Thus, this term alone would tend to make  $I_{sv}$  large. However, we have the competing factor that at low altitude angles atmospheric depletion is greatest.

To summarize on Dr. Ghai's question, in general a minimum incidence occurs on or about December 22, but it depends upon the latitude and the orientation of the surface.

**AUTHOR'S CLOSURE (Doctor Jordan):** Professor Kayan has shown interest in the statement that 40 percent of the energy requirements of this country are involved in space or comfort heating. In earlier conversations with Professor Kayan he has also questioned whether or not this figure includes energy requirements for transportation purposes.

There are several sources of information varying somewhat in the exact figure, but the same in general magnitude. One of the best of these references is the book *Energy in the Future* by Putnam, prepared in his capacity as a consultant to the Atomic Energy Commission. This book was published in 1953 and provides detailed substantiating evidence on all assumptions.

I am sure that some of Mr. Putnam's figures will be of interest to you. In the year 1800 about 92 percent of the energy requirements in this country were consumed in space heating, about 8 percent in process heating, and almost none in work. In the year 1930 the comfort or space heating requirements had dropped to approximately 40 percent and process heating to about 10 percent. The remaining 50 percent was consumed in work. The space or comfort heating fraction includes that part used in the heating of transportation facilities. Energy required for the transportation of vehicles such as automobiles, trains and airplanes, was allotted to the work fraction.

Currently, the space and comfort heating requirements are approximately 35 to 40 percent of the entire energy requirements of this country and it is projected that the energy-use curve will approach 30 percent by the year 2000. Despite the changing fractions of all energy requirements, the total consumed in each category has increased greatly over the years. While the 30 percent space heating figure may stabilize, the total energy requirement for space heating will increase greatly during the last half century as a result of the booming population. This is the reason why the vast potential availability of solar energy becomes so interesting, since it is at the space and comfort heating levels that we can probably best first utilize solar energy.



**1534**

## PATHS OF HORIZONTALLY PROJECTED HEATED AND CHILLED AIR JETS

By ALFRED KOESTEL\*, CLEVELAND, OHIO

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SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.,  
in cooperation with Case Institute of Technology, Cleveland, Ohio

**T**HE PATHS assumed by horizontally projected heated and chilled air jets under the influence of buoyant forces are of practical importance when heating or cooling with outlets located on walls. When the path of a chilled air jet is such that it prematurely enters the occupied zone, considerable draft may be experienced especially if the temperature differences and velocities are still high due to insufficient mixing.

Equation 2 gives the path or trajectory of horizontally projected heated or chilled air jets. It also defines the path in terms of the drop or rise of the jet centerline velocity axis for various distances from the discharge outlet. A typical path of a chilled air jet is shown schematically in Fig. 1. The trajectory equation is compared with test data obtained by various experimenters in order to substantiate the assumptions made in the mathematical derivation of Equation 2. The detailed mathematical analyses appears in the Appendix.

### BASIC EQUATIONS

From the analysis presented in the Appendix and based on the simplifying assumptions already stated, the following differential equation for the trajectory of a horizontally projected non-isothermal air jet is derived as the equation:

$$\frac{(\Delta t_o \beta g D_o / V_o^2) [(a/b + 1) / K D_o^2] [1 + (dY/dX)^2]^{1/2}}{\int_0^x [1 + (dY/dX)^2]^{1/2} dX} = \pm d^2 Y / dX^2 \quad \dots \dots (1)$$

By assuming that the slope of the jet trajectory is not too great, an approximate

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Presented at the 61st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1955.

solution to Equation 1 can be made (as developed in the Appendix), which results in the equation:

$$(\Delta t_o \beta g D_o / V_o^3) [(a/b + 1)/6K] (X/D_o)^3 = \pm (Y/D_o) \quad \dots (2)$$

Equation 2 approximately defines the path of the centerline axis of the jet as illustrated in Fig. 1. The centerline of the jet path is the line of maximum air velocity and also of maximum air temperature difference.

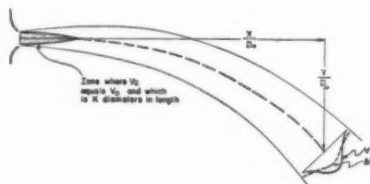


FIG. 1. DIAGRAM OF THE PATH ASSUMED BY A CHILLED AIR JET DISCHARGING HORIZONTALLY FROM A NOZZLE

Also as developed in the Appendix, the ratio  $a/b$  in Equation 2 is a function of the overall effective turbulent Prandtl number of the jet as defined by the equation:

$$b/a = 4/[1 + (1/N_{Prt})] - 1. \quad \dots (3)$$

A Prandtl number of 0.7 has been suggested for non-isothermal jets by such experimenters as Nottage<sup>1</sup>, Forstall and Shapiro<sup>2</sup>, and Corrsin and Uberoi<sup>3</sup>.

Whether the sign is plus or minus in Equation 2 is determined by the fact that the vertical displacement in diameters,  $Y/D_o$ , of the jet centerline axis is the *drop* when the temperature difference  $\Delta t_o$  is for a chilled jet, and the *rise* when the temperature difference,  $\Delta t_o$ , is for a heated jet.

The dimensionless parameter,  $V_o^2/\beta \Delta t_o g D_o$ , in Equation 2 is the ratio of the momentum force to the buoyant force or gravity force evaluated at the outlet.

An analytical comparison between the trajectory equation of a water jet in air and a heated or chilled air jet in air may provide an insight into the general nature of the paths assumed by fluid jets under the influence of gravity forces. The dimensionless parameter  $V_o^2/\beta \Delta t_o g D_o$ , which is the ratio of momentum or inertia force to the buoyant or gravity forces, can also be represented by the conventional Froude number.\* The analysis is given in the Appendix.

The constant  $K$  in Equation 2 is the familiar constant of proportionality for the centerline velocity variation of an isothermal jet as defined by the equation:

$$V_e/V_o = K/(X/D_o) \quad \dots (4)$$

$K$  also represents the apparent length of the constant-velocity zone occurring near the face of the outlet, for when  $V_e/V_o$  equals unity  $K$  becomes equal to  $X/D_o$ , the zone length in diameters.

The constant  $K$  has been verified experimentally by Tuve<sup>4</sup> and others and has

<sup>1</sup> Exponent numerals refer to References.

\* The parameter,  $V_o^2/\beta \Delta t_o g D_o$ , is similar to the dimensionless parameter called the Froude number in hydraulics. The Froude number is equal to the square root of the ratio of the momentum force to the gravitational force in a fluid.

been found to vary somewhat with outlet velocity, with the percentage free area of the outlet (if covered with a grid-type lattice), with the type of approach to the outlet, and with the amount of turbulence induced behind the outlet. Differences between the jet density and the density of the receiving medium affect the values of  $K$  only when the jet temperature difference,  $\Delta t_o$ , is excessively large. For outlet velocities above 1000 fpm and diameters greater than 6 in., values for  $K$  of 6.0 to 6.5 are representative of many tests on nozzles. Experimental indications are that  $K$  is a function of the Reynolds number (evaluated at the discharge nozzle) only in the lower ranges, and that for a Reynolds number greater than about  $3 \times 10^4$ , values of  $K$  will be greater than 6.0. If the apparent point source of the jet as determined by the angle of expansion of the velocity profile is located on the face of the nozzle, the value of  $K$  in Equation 2 based on available test data can be expected to range from 6.5 down to approximately 4.0 depending on the Reynolds number evaluated at the jet outlet.

### METHOD OF ANALYSIS

The jet trajectory is analyzed in terms of principles involving conservation of thermal energy and the equating of the buoyant forces to the change in jet momentum. The differences in rates of momentum transfer and heat transfer are taken into account by the use of an overall effective turbulent Prandtl number. Available experimental knowledge of free air jet behavior is used wherever possible to evaluate constants in the derived equations.

An equation describing the path of horizontally projected non-isothermal jets can be formulated if the following simplifying assumptions are made:

1. The absolute value of the air density is constant throughout the jet, but the differences in density are taken into account when the buoyant forces are considered. This assumption is valid if the temperature differences between the jet and the surroundings are not too great. Temperature differences normally involved in the application of jets to space heating or cooling can be considered not too great.
2. The velocity and temperature-difference profiles in the fully developed region of the jet can be represented by an error-function type curve or normal probability curve. This has been found experimentally to be a satisfactory assumption.
3. The difference between the temperature-difference profiles and velocity profiles at a given section of the jet is due to the difference between heat and momentum diffusion in a turbulent jet. This difference in diffusion rate can be expressed by the use of an overall effective turbulent Prandtl number.
4. The rate of spread or expansion of the temperature-difference and velocity profiles are constant along the curved trajectory of the air jet, but differ in value due to the difference between heat diffusion and momentum diffusion.
5. The velocity and temperature-difference profiles remain symmetrical about the centerline of the curved path of the air jet. See Fig. 1.
6. Thermal energy (or enthalpy flow) is conserved along the curved path of the air jet.
7. In order to mathematically solve Equation 1, which is the differential equation of the jet trajectory, and to derive Equation 2, an assumption was necessary that the slope of the jet trajectory is not too great.

### COMPARISON WITH TEST DATA

In Fig. 2, Equation 2 is compared with test data obtained by Nelson and Stewart<sup>8</sup> on chilled air projected into a room from outlets of various dimensions. Equa-

tion 2 is plotted for a value of  $a/b + 1$  equal to 2.54 which corresponds to an overall effective turbulent Prandtl number of 0.7 in Equation 3. A value for  $K$  of 6.5 was selected for the plot of Equation 2 in Fig. 2. With a  $N_{Pr}$  equal to 0.7 and a  $K$  of 6.5 Equation 2 becomes:

$$0.065 (\beta \Delta t_{og} D_o / V_o^2) (X/D_o)^3 = \pm (Y/D_o) \dots \dots \dots (5)$$

which is the equation plotted in Fig. 2.

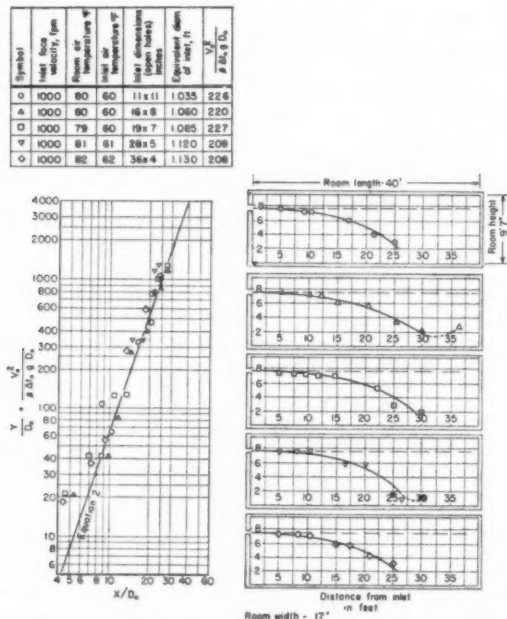


FIG. 2. CENTERLINE TRAJECTORIES OF CHILLED AIR JETS IN A ROOM FROM RECTANGULAR OPEN HOLES COMPARED WITH EQUATION 2 (FROM DATA REPORTED BY D. W. NELSON AND D. J. STEWART<sup>5</sup>)

The test points of Nelson and Stewart lying on the termination of the dotted portion of the curves defining the jet path as shown in Fig. 2 are not compared on the plot with Equation 2 because the chilled air jet paths had already been influenced by the floor. The paths of the air jets shown in Fig. 2 were determined by means of smoke and by velocity traverses made with a deflecting vane type anemometer.

In Fig. 3, Equation 2 is compared with test data obtained by Van Alsbarg<sup>6</sup> on the trajectories of heated and chilled air jets from a 12- x 6-in. grille. Note the relatively low momentum force and large buoyant force ( $V_o^2 / \beta \Delta t_{og} D_o = 3.84$ ) for the trajec-

tory defined by curve "a". The same values for  $K$  and  $N_{Prt}$  were used in plotting Equation 2 in Fig. 3 as in Fig. 2. The paths of the air jets shown in Fig. 3 were determined by means of smoke discharged through the outlet. In his tests Van Alsburg also determined the path of an isothermal jet for comparison with the paths of the heated and chilled jets. The slight downward path of the isothermal jet shown in Fig. 3 is probably caused by room air currents external to the jet, and serves to emphasize the effect of extraneous disturbances on the trajectories of jets.

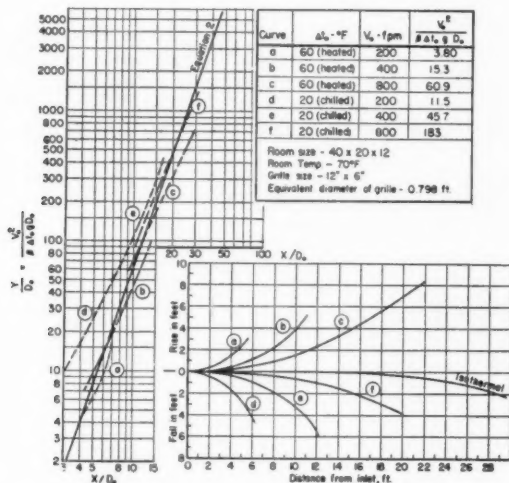


FIG. 3. CENTERLINE TRAJECTORIES OF HEATED AND CHILLED AIR JETS FROM A 12 x 6 IN. GRILLE COMPARED WITH EQUATION 2 (FROM DATA REPORTED BY J. H. VAN ALSBURG<sup>6</sup>)

In Fig. 4, Equation 2 is compared with test data obtained by Greenlaw and Hart<sup>7</sup> on chilled air jet paths from a 24- x 6-in. grille for different outlet velocities and for a temperature difference of 15 deg below room temperature. The agreement between test data and Equation 2 is good even for the lower outlet velocities. The same values for  $K$  and  $N_{Prt}$  were used in plotting Equation 2 in Fig. 4 as in Fig. 2. The paths of the air jets shown in Fig. 4 were determined by means of smoke discharged through the outlet, and were checked by probing for the edge of the jet by a direct reading anemometer.

From the tests made by Greenlaw and Hart,<sup>7</sup> they conclude that neither grille aspect ratio, nor breaking up the air stream into individual jets at the grille, nor converging the individual jets had any effect on the drop of the air stream.

In Fig. 5 Equation 2 is compared with test data obtained by Nottage<sup>1</sup> on the path of the centerline velocity axis of a chilled air jet from a 6 in. nozzle with an outlet velocity of 500 fpm and a temperature difference of 39 deg below room tempera-

ture. The same values for  $K$  and  $N_{Prt}$  were used in plotting Equation 2 in Fig. 5 as in Fig. 2. Nottage's test data indicate a parabolic trajectory or a slope of two for the test points in Fig. 5 compared to a slope of three according to Equation 2. The path of the chilled air jet centerline velocity was determined by means of velocity traverses made by a specially designed heated thermocouple anemometer.<sup>1</sup>

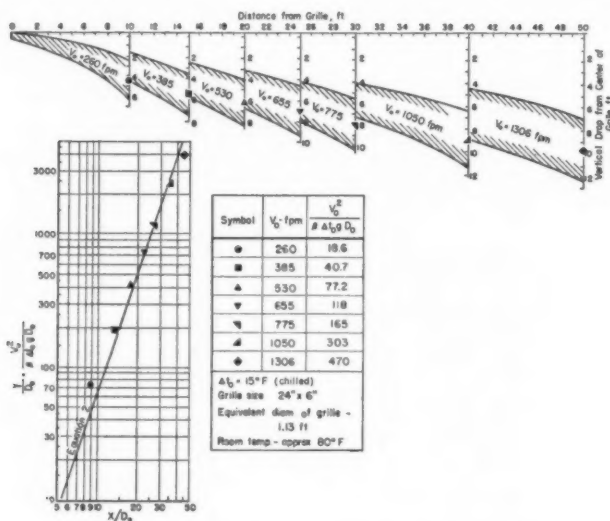


FIG. 4. CHILLED AIR JET TRAJECTORIES FROM A 24- X 6-IN. GRILLE COMPARED WITH EQUATION 2 (FROM DATA REPORTED BY A. L. GREENLAW AND T. S. HART<sup>7</sup>)

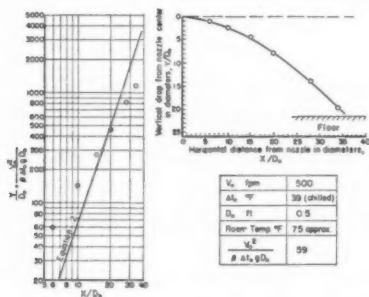


FIG. 5. CENTERLINE TRAJECTORY OF A CHILLED JET FROM A 6-IN. NOZZLE COMPARED WITH EQUATION 2 (FROM DATA REPORTED BY H. B. NOTTAGE<sup>1</sup>)

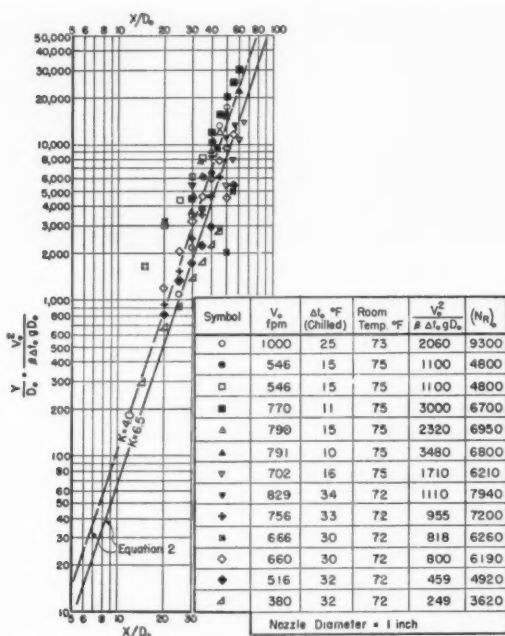


FIG. 6. CENTERLINE TRAJECTORIES OF CHILLED JETS FROM A 1-IN. NOZZLE COMPARED WITH EQUATION 2 (FROM DATA OBTAINED AT CASE INSTITUTE OF TECHNOLOGY)

In Fig. 6, Equation 2 is compared with test data obtained at Case Institute of Technology on the path of chilled air jets discharged from a 1-in. nozzle at different velocities and temperatures. Note that Equation 2 is plotted in Fig. 6 for two different values of  $K$  but for an overall effective turbulent Prandtl number of 0.7 in both cases. The paths of the chilled air jets were made visible by titanium tetrachloride smoke generated in a plenum chamber on which the nozzle was mounted. A photograph was taken of the smoke pattern and the trajectory coordinates were visually measured on this photograph, then plotted in Fig. 6. When the drop in diameters of the centerline velocity axis of the jet is about 5, an error of  $\pm 1$  diameter is possible when visually evaluating the centerline locus of the smoke pattern from the photograph. In addition, errors may result from the fact that the diffusion of smoke in an air jet involves the turbulent Schmidt number which may result in some discrepancy between the centerline of the smoke path and the path of the maximum centerline velocity axis. Fig. 7 is a typical photograph of a chilled air jet path.

The path of the chilled air jets emanating from a 1-in. nozzle proved to be erratic and to be very susceptible to ambient room air disturbances since the magnitude of

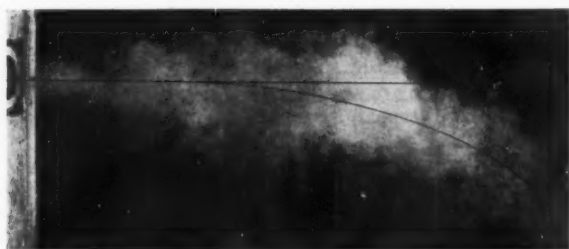


FIG. 7. SMOKE PATH OF A CHILLED AIR JET FROM A 1-IN. NOZZLE SHOWING ESTIMATED MAXIMUM VELOCITY CENTERLINE AXIS. ( $\Delta t_o = 15$  F;  $V_o = 546$  FPM)

the disturbing force is appreciable contrasted to the relatively small momentum force of the discharging air jet. For large nozzles the ambient disturbing forces could be minimized. Curtains were hung in the vicinity of the air jet to reduce the effect on the jet trajectory of air currents that normally exist in a large laboratory. Fig. 7 is an instantaneous picture of a fluctuating air jet trajectory, and therefore, the data plotted in Fig. 6 should be statistically evaluated. By visual examination of the data in Fig. 6, the curve of Equation 2 for a  $K$  of 4 seems to be the better statistical representation of the data. The agreement of the data with this lower  $K$  value is to be expected, due to the relatively low Reynolds numbers involved in air jets from a 1-in. nozzle. Note that the tabulated Reynolds numbers in the legend of Fig. 6 are low contrasted to the Reynolds number values ( $3 \times 10^4$ ) that should be realized for  $K$  values greater than 6.

Data obtained at Case on the rise of horizontally projected heated jets from a 2-in. nozzle are presented in Figs. 8 and 8a. The data were evaluated from photographs of smoke patterns using a procedure similar to the evaluation of the data presented in Fig. 6. In Fig. 8 it can again be noted that the data tends to group around a  $K$  value which is lower than 6.5 due to low Reynolds numbers at the jet discharge. (See Fig. 8a for tabulated values of Reynolds numbers). Even though considerable scatter exists for the data shown in Fig. 8, the trend is for the test points to band with a spread of approximately  $\pm 50$  percent around some estimated mean curve through the data. Fig. 9 shows a typical photograph of the smoke path of a heated jet.

In Fig. 10 is shown a summary of the test data on the heated and chilled jet trajectories presented in Figs. 6 and 8, and an envelope encompassing all of the test points in Figs. 6 and 8 is illustrated and compared with Equation 2 for two values of  $K$  and for a turbulent Prandtl number of 0.7.

The author believes that certain refinements in experimental technique and test equipment would have reduced the spread in the experimental test data shown in Figs. 6 and 8 to approximately  $\pm 25$  percent.

#### PRACTICAL EXAMPLE

As a simple example in finding the drop of a horizontally projected chilled air jet, consider a 1 ft diameter nozzle discharging chilled air with a temperature difference

of 20 deg and with a velocity of 1000 fpm. The room air temperature is 75 F. It is required (a) to find the drop of the maximum centerline velocity axis at a distance of 30 ft from the nozzle, and (b) to determine the drop of the lower edge of the chilled air jet also at a distance of 30 ft.

The ratio of momentum to buoyancy is

$$V_o^2 / \Delta \rho_o \beta g D_o = [(1000/60)^2 \times (460 + 75)] / (20 \times 32.2 \times 1) = 230$$

Assuming a  $K$  equal to 6.5 and a  $N_{Prt}$  equal to 0.7 the drop of the maximum centerline velocity axis computed from Equation 2 or Equation 5 is:

Solution (a):

$$0.065 \times 1/230 \times (30/1)^3 = Y/1$$

$$Y = 7.7 \text{ ft}$$

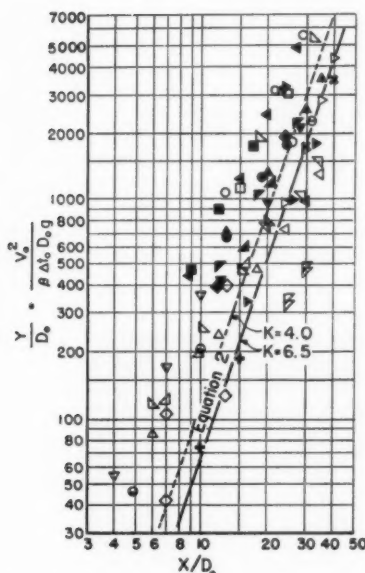


FIG. 8. CENTERLINE TRAJECTORIES OF HEATED AIR JETS FROM A 2-IN. NOZZLE COMPARED WITH EQUATION 2 (FROM DATA OBTAINED AT CASE INSTITUTE OF TECHNOLOGY)

Symbol	$V_o$ (ft/min)	$\Delta T_o$ (heated)	$\frac{V_o^2}{\beta \Delta \rho_o g D_o}$	$(N_{R})_o$
▷	658	22	547	10740
◁	600	38	264	9820
▽	426	45	112	6960
+	518	40	187	8470
×	834	22	879	13620
●	1320	21	2300	21600
○	1380	20	2650	22600
▽	605	20	510	9910
●	720	20	720	11770
◁	212	14	90	3470
△	215	20	64	3510
△	216	26	50	3530
▽	217	31	42	3550
◇	217	38	38	3550
○	219	44	31	3590
□	1030	28	1043	16780
○	1040	27	1108	17000
◁	1030	27	1090	16800
▲	861	34	606	14100
▲	654	25	476	10700
▽	654	24	495	10700
◆	660	31	390	10800
■	748	37	419	12200
●	750	41	381	12250
▶	756	47	338	12350
▶	870	52	404	14220
▽	875	56	381	14320

Room temperature 78°F

FIG. 8a. LEGEND FOR HEATED AIR JETS FROM A 2-IN. NOZZLE (DATA OBTAINED AT CASE INSTITUTE OF TECHNOLOGY)

*Solution (b):* Since the jet boundary expands with a natural free angle of about 20 deg<sup>4</sup>, the distance of the lower edge of the chilled jet from the maximum centerline velocity axis will be approximately (assuming that the slope of the trajectory is not too great),

$$30 \times \tan 20^\circ/2 = 5.3 \text{ ft}$$

The drop of the lower edge of the chilled air jet is therefore,

$$5.3 + 7.7 = 13 \text{ ft}$$

#### CONCLUSIONS AND RECOMMENDATIONS

Equation 2 indicates the important variables for exploring the trajectories of horizontally projected non-isothermal jets. Equation 2 has been compared with test data for ratios of momentum to buoyancy ( $V_o^2/\beta\Delta t_o g D_o$ ) ranging from about 4 to 4000.

The validity of the trajectory equation has been checked to an approximation by experimental results, but suggestions are made for additional experimental work to refine the constants in the equation and to reveal additional variables. The agreement between the analyses and the data in its present form does not necessarily give a broad validation of either the data or the theory.

No attempt has been made to estimate the coefficients of discharge or the effective free areas of the grilles and outlets presented in this paper or to include these factors in the test points when comparisons are made with Equation 2. It is understood that the neglecting of the effective free areas and the coefficients of discharge would account for some of the deviation of test data from Equation 2.

Simplifying assumptions have been made in the analytical treatment of the jet problem, and the further experiment is necessary to indicate the seriousness of actual deviations from each individual assumption.

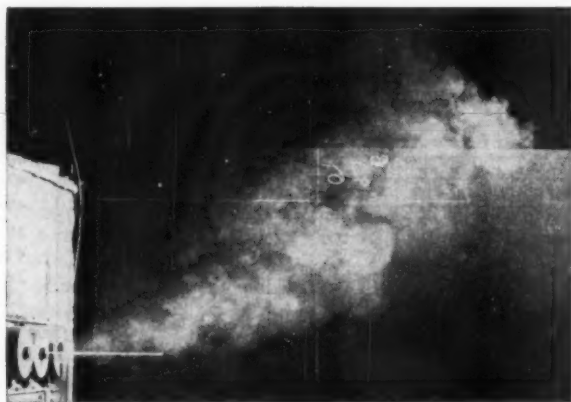


FIG. 9. SMOKE PATH OF A HEATED AIR JET FROM A 2-IN. NOZZLE. ( $\Delta t_o = 26 \text{ F}$ ;  $V_o = 216 \text{ FPM}$ )

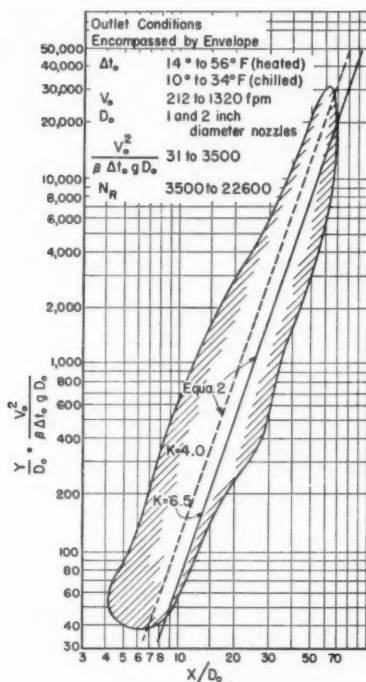


FIG. 10. ENVELOPE CONTAINING ALL DATA ON HEATED AND CHILLED TRAJECTORIES OF FIGS. 6 AND 8 COMPARED WITH EQUATION 2

Equation 2 is recommended for engineering use, pending its refinement by more rigorous experimental work. Equation 2 is not to be considered exact until full experimental verification is accomplished and any limitation on the empiricism determined.

#### ACKNOWLEDGMENTS

Grateful acknowledgment is made of the financial support from the Designers for Industry, Inc. Research Foundation, Cleveland, Ohio. The author is indebted to Prof. G. L. Tuve for his help and suggestions.

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## APPENDIX

### HYDRAULIC ANALOGY

An analytical comparison between the trajectory equation of a water jet in air and a heated or chilled air jet in air may provide an insight into the general nature of the paths assumed by fluid jets under the influence of gravity forces.

For a water jet discharged horizontally into air the trajectory is parabolic and is defined by the following equation which can be found in any standard text on fluid mechanics:

$$\frac{X^2}{4 \left( \frac{V_o^2}{2g} \right)} = Y \quad \dots \quad (A-1)$$

If the distances  $X$  and  $Y$  are expressed in terms of diameters, Equation A-1 can be restated as follows:

$$\frac{1}{2} \frac{D_o g}{V_o^2} \left( \frac{X}{D_o} \right)^2 = \frac{Y}{D_o} \quad \dots \quad (A-2)$$

In hydraulics the square root of the ratio of momentum or inertia forces to gravity forces is called the Froude Number and is written as follows:

$$N_F = \frac{V}{\sqrt{gL}} \quad \dots \quad (A-3)$$

where  $L$  is a characteristic length and is usually the depth of the liquid in the case of open channel flow. For a water jet discharged from a nozzle into ambient air, the characteristic length,  $L$ , can be taken as the diameter of the nozzle; therefore, Equations A-2 and A-3 can be combined to obtain the following equation:

$$\frac{1}{2} \frac{1}{N_F^2} \left( \frac{X}{D_o} \right)^2 = \frac{Y}{D_o} \quad \dots \quad (A-4)$$

Equation A-4 states that the path of a water jet in air depends on the Froude Number evaluated at the nozzle.

Equation 2 derived for an air jet can be transformed into an equation similar to Equation A-4. The dimensionless parameter  $\frac{V_o^2}{\Delta t_o \beta g D_o}$ , which is the ratio of momentum or inertia force to the buoyant or gravity forces, can also be represented by the Froude number; therefore, Equation 2 becomes:

$$\frac{1}{N_F^2} \frac{(a/b + 1)}{oK} \left( \frac{X}{D_o} \right)^2 = \frac{Y}{D_o} \quad \dots \quad (A-5)$$

Since both the heat and momentum diffusion of a jet of heated or chilled water into air might be considered negligible, the Prandtl number for this case becomes unity and the ratio  $a/b$  computed from Equation 3 becomes equal to one. Also the length of the apparent constant velocity zone,  $K$  in Equation 2, or the zone of potential flow extends indefinitely for a water jet in air due to negligible mixing of the two fluids. If the trajectory slope is not too great,  $K$  becomes approximately equal to  $X/D_o$  and Equation A-5 now becomes,

$$\frac{1}{3} \frac{1}{N_F^2} \left( \frac{X}{D_o} \right)^2 = \frac{Y}{D_o} \quad \text{. . . . . (A-6)}$$

which is approximately the same as Equation A-4 derived for a water jet only. The comparison between Equations A-4 and A-6 provides an indication that the general nature of the results presented in this paper may have utility in other technical fields such as meteorology, oceanography, and hydraulics.

Since one of the constants in Equation 2, namely  $K$ , seems to vary with the Reynolds number evaluated at the outlet, one might be inclined to generalize the trajectory equation into the following dimensionless form:

$$\frac{Y}{D_o} = \mathcal{F} \left( \frac{X}{D_o}, N_R, N_{Pr}, N_F \right) \quad \text{. . . . . (A-7)}$$

where  $\mathcal{F}$  expresses a general function. The dimensionless variables in Equation A-7 may form the basis of a more exact trajectory equation if the form of equation is determined by experimentation.

#### MOMENTUM ANALYSIS

A typical path assumed by a horizontally projected chilled air jet under the influence of buoyant forces is schematically illustrated in Fig A-1.

The total jet momentum acting tangent to the centerline velocity axis at any distance from the nozzle is,

$$M_S = 2\pi\rho \int_{r=0}^{r=\infty} V^2 r dr \quad \text{. . . . . (A-8)}$$

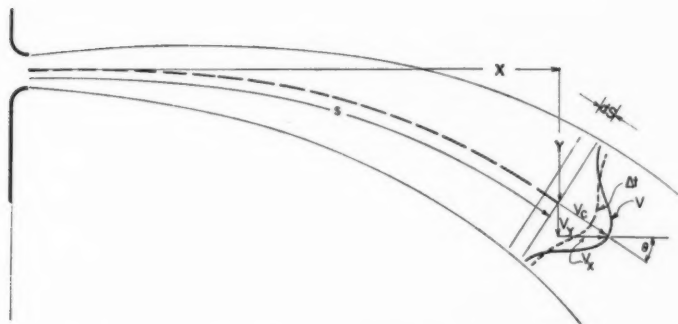


FIG. A-1. DIAGRAM OF THE PATH ASSUMED BY A CHILLED AIR JET DISCHARGING HORIZONTALLY FROM A NOZZLE

Pressure difference forces are considered negligible in the free jet. Fig. A-1 is the diagram to which Equation A-8 is applied.

A normal probability-type velocity distribution can be assumed for the jet-velocity profile having the form of,

$$V = V_e e^{-ar^2} \quad (A-9)$$

where the factor  $a$  can be called a shape factor since its value determines the width of the profile for a given value of the centerline velocity,  $V_e$ .

If the function of  $V$  in Equation A-9 is substituted into Equation A-8 and the integration is accomplished, the following equation is obtained:

$$M_s = \frac{\pi \rho V_e^2}{2a} \quad (A-10)$$

At any distance from the nozzle, the component of the total jet momentum,  $M_s$ , in the  $X$  direction is,

$$M_X = 2\pi \rho \cos \theta \int_{r=0}^{r=\infty} V^2 r dr \quad (A-11)$$

and the component of the total jet momentum,  $M_s$ , in the  $Y$  direction is,

$$M_Y = 2\pi \rho \sin \theta \int_{r=0}^{r=\infty} V^2 r dr \quad (A-12)$$

The ratio,  $M_Y/M_X$ , of the jet momentum is,

$$\frac{M_Y}{M_X} = \frac{2\pi \rho \sin \theta \int_{r=0}^{r=\infty} V^2 r dr}{2\pi \rho \cos \theta \int_{r=0}^{r=\infty} V^2 r dr} = \frac{\sin \theta}{\cos \theta} = \tan \theta = \frac{dY}{dX} \quad (A-13)$$

#### HEAT FLOW ANALYSIS

If it is assumed that the temperature-difference profile in a non-isothermal air jet can also be defined by a normal probability curve, we have,

$$\Delta t = \Delta t_e e^{-br^2} \quad (A-14)$$

where  $b$  is the shape factor for the temperature-difference profile. ( $b$  will not have the same value as  $a$  in Equation A-9 due to the difference in rates of momentum and heat diffusion.)

The heat flow or enthalpy flow in a direction tangent to the centerline velocity axis at any distance from the nozzle is,

$$Q_s = \gamma C_p 2\pi \int_{r=0}^{r=\infty} \Delta t V r dr \quad (A-15)$$

If the function of  $V$  in Equation A-9 and the function of  $\Delta t$  in Equation A-14 are substituted into Equation A-15 and the integration accomplished, the following equation is obtained:

$$Q_s = \frac{\gamma C_p \pi \Delta t_e V_e}{a + b} \quad (A-16)$$

The enthalpy flow at the nozzle discharge is,

$$Q_o = \frac{\pi D_o^2}{4} V_o \Delta t_o \gamma C_p \quad (A-17)$$

If enthalpy is conserved, Equations A-16 and A-17 can be equated, and we have,

$$\frac{\pi D_o^3}{4} V_o \Delta t_o \gamma C_p = \frac{\gamma C_p \pi \Delta t_o V_e}{a + b}$$

and

$$\Delta t_o = \frac{D_o^3 V_o \Delta t_o (a + b)}{4 V_e} \dots \dots \dots (A-18)$$

#### BUOYANT FORCE ANALYSIS

The buoyant force at any cross-section of a non-isothermal jet can be determined from the assumed temperature-difference profile, *i.e.*, Equation A-14. The buoyant force per unit volume for a gas is  $\gamma \beta \Delta t$  where  $\beta$  is the coefficient of expansion and is equal to  $1/T_a$  for a perfect gas.

The buoyant force  $F$  at a given stream cross-section  $dS$  in length is,

$$F = dS \beta \gamma 2\pi \int_{r=0}^{r=\infty} \Delta t r dr \dots \dots \dots (A-19)$$

If the function of  $\Delta t$  in Equation A-14 is substituted into Equation A-19 and the integration accomplished, we have,

$$F = \frac{(\Delta t_o \beta \gamma \pi) dS}{b} \dots \dots \dots (A-20)$$

#### SHAPE FACTORS $a$ AND $b$ AS A FUNCTION OF THE PATH, $S$

Assuming a uniform expansion of the air jet along its trajectory in the  $S$  direction, a ratio  $V_1/V_o$  measured along an arbitrary expansion line drawn through  $r_1$  at  $V_1$  becomes independent of  $X$ . The ratio  $V_1/V_o$  could be taken as, say, one half. This is also the condition of dynamic similarity of the velocity profiles at any point along the jet path. Also,

$$\frac{dr_1}{dS} = \text{Constant}, \dots \dots \dots (A-21)$$

and Equation A-9 can be written as

$$V_1/V_o = \text{Constant} = e^{-ar_1^2},$$

and

$$ar_1^2 = \text{Constant} \dots \dots \dots (A-22)$$

Integrating Equation A-21 we have,

$$r_1 = S \text{ Constant} \dots \dots \dots (A-23)$$

where the constant of integration is zero. Combining equations A-23 and A-22 we have,

$$a = \frac{C_2}{S^2} \dots \dots \dots (A-24)$$

when  $C_2$  is a constant.

A similar analysis on the temperature-difference profile gives,

$$b = \frac{C_1}{S^2} \dots \dots \dots (A-25)$$

where  $C_1$  is a constant.

The apparent point sources for the velocity profile and for the temperature-difference profile are assumed to lie on the face of the outlet so that both of the path lengths,  $S$ , in Equations A-24 and A-25 are the same.

#### DERIVATION OF EQUATION

For horizontally projected non-isothermal jets the  $Y$  component of the total jet momentum changes due to the action of the buoyant forces. An equation in the form of  $F = \pm d(M_Y)$  can be written, and expressions already derived for the buoyant force and momentum substituted. The plus or minus sign indicates an increase or decrease in momentum depending on the direction of the buoyant force or whether the jet is heated or chilled.

The  $X$  component of the total momentum is conserved since no external forces are assumed to act in the horizontal direction; therefore,  $M_X = M_o$ .

Since  $F = \pm d(M_Y)$ , by means of Equation A-13,

$$F = \pm d(M_X \tan \theta) = \pm M_X d\left(\frac{dY}{dX}\right). \quad (A-26)$$

Also  $M_o = \frac{\pi D_o^2}{4} \rho V_o^2 = M_X$ , which can be substituted into Equation A-26, to give,

$$F = \pm \frac{\pi D_o^2}{4} \rho V_o^2 d\left(\frac{dY}{dX}\right). \quad (A-27)$$

Substituting Equation A-26 into Equation A-27, we have

$$\frac{(\Delta t_e \beta \gamma \pi) dS}{b} = \pm \frac{\pi D_o^2}{4} \rho V_o^2 d\left(\frac{dY}{dX}\right). \quad (A-28)$$

Eliminating  $\Delta t_e$  from Equation A-28 by substitution of Equation A-18, we have

$$\frac{\Delta t_o(a/b + 1)\beta g dS}{V_o V_e} = \pm d\left(\frac{dY}{dX}\right). \quad (A-29)$$

In Equation A-29,  $V_e$  can be expressed in terms of  $X$  and  $Y$  by the following manipulation:

$$\text{Since } M_X = M_o \cos \theta, \text{ one can } \quad (A-30)$$

substitute Equation A-10 into Equation A-30, thus,

$$M_X = \frac{\pi \rho V_o^2}{2a} \cos \theta. \quad (A-31)$$

Since  $M_X = M_o$ ,

$$\frac{\pi D_o^2}{4} \rho V_o^2 = \frac{\pi \rho V_e^2}{2a} \cos \theta$$

and

$$V_e = \frac{D_o V_o}{2} \sqrt{\frac{2a}{\cos \theta}}. \quad (A-32)$$

By substituting Equation A-24 into Equation A-32, and noting that,

$$\cos \theta = \frac{dX}{dS} \quad (A-33)$$

and

$$dS = \sqrt{(dX)^2 + (dY)^2} = \sqrt{1 + \frac{dY^2}{dX}} dX$$

or

$$S = \int_0^X \sqrt{1 + \frac{dY^2}{dX}} dX \quad \dots \quad (A-34)$$

which also can be substituted into Equation A-3, thus,

$$V_e = \frac{D_o V_o}{2} \frac{(C_2)^{\frac{1}{2}} \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}}}{\int_0^X \left[ 1 + (dY/dX)^2 \right]^{\frac{1}{4}} dX} \quad \dots \quad (A-35)$$

Let  $C_2$  equal  $2K_p^2$  according to the analysis presented in the Appendix of a previously published paper\*, and substituting this value into Equation A-35, we have,

$$V_e = \frac{D_o V_o K_p \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}}}{\int_0^X \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}} dX} \quad \dots \quad (A-36)$$

Substituting Equation A-36 into Equation A-29, we have,

$$\frac{\Delta t_o (a/b + 1) \beta g dS \int_0^X \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}} dX}{V_o^2 D_o K_p \left[ 1 + (dY/dX)^2 \right]^{\frac{1}{4}}} = \pm d \left( \frac{dY}{dX} \right) \quad \dots \quad (A-37)$$

Substituting Equation A-34 for  $dS$ , we have

$$\frac{\Delta t_o (a/b + 1) \beta g}{V_o^2 D_o K_p} \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}} \int_0^X \left[ 1 + (dY/dX)^2 \right]^{\frac{1}{4}} dX = \pm \frac{d}{dX} \left( \frac{dY}{dX} \right) \quad \dots \quad (A-38)$$

The left term of Equation A-38 can be rearranged to obtain the following form of equation,

$$\left( \frac{\Delta t_o \beta g D_o}{V_o^2} \right) \left( \frac{a/b + 1}{D_o^2 K_p} \right) \left[ 1 + \left( \frac{dY}{dX} \right)^2 \right]^{\frac{1}{4}} \int_0^X \left[ 1 + (dY/dX)^2 \right]^{\frac{1}{4}} dX = \pm \frac{d^2 Y}{dX^2} \quad \dots \quad (A-39)$$

The term  $\frac{\Delta t_o \beta g D_o}{V_o^2}$  is a dimensionless ratio of the buoyant or gravity forces to the momentum or inertia forces at the nozzle. The term  $a/b$  is a function of the over-all effective turbulent Prandtl number defined in the earlier paper as follows:

$$b/a = \frac{4}{1 + \frac{1}{N_{Prt}}} - 1$$

$K_p$  is the length of the constant velocity zone near the face of the outlet measured from the apparent point source as defined. Also according to the previous paper, if the apparent point source is assumed to lie on the face of the outlet,  $K_p$  becomes equal to  $K$ . Experimental indications are that  $K$  is a function of the Reynolds number for low values only. From tests on nozzles  $K$  is approximately 6.0 to 6.5 for the higher values of the Reynolds number. Assuming the apparent point source as determined

\* ASHVE RESEARCH REPORT NO. 1512—Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air, by Alfred Koestel (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 385).

by the velocity profile of the jet to lie on the face of the discharge nozzle, Equation A-39 can be restated as follows with  $K$  replacing  $K_p$ ,

$$\left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) \frac{(a/b + 1)}{K} \left(\frac{1}{D_o^2}\right) \left[1 + \left(\frac{dY}{dX}\right)^2\right]^{\frac{1}{2}} \int_0^X [1 + (dY/dX)^2]^{\frac{1}{2}} dX = \pm \frac{d^2 Y}{dX^2} \quad (\text{A-40})$$

In order to derive a trajectory equation from Equation A-40, an approximate solution may be obtained if the assumption is made that the slope of the trajectory  $dY/dX$  or  $\tan \theta$  is not too great. This assumption makes the term  $[1 + (dY/dX)^2]^{\frac{1}{2}}$  and  $[1 + (dY/dX)^2]^{\frac{1}{2}}$  in Equation A-40 approximately equal to unity.

With this approximation, Equation A-40 becomes,

$$\left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) \frac{(a/b + 1)}{K} \left(\frac{1}{D_o^2}\right) \int_0^X dX = \pm \frac{d^2 Y}{dX^2} \quad \dots \quad (\text{A-41})$$

which can be readily solved by integration.

Performing the integration we have,

$$\left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) \frac{(a/b + 1)}{K} \left(\frac{1}{D_o^2}\right) X = \pm \frac{d^2 Y}{dX^2} \quad \dots \quad (\text{A-42})$$

Integrating Equation A-42 again, we have,

$$\left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) \frac{(a/b + 1)}{K} \frac{1}{D_o^2} \frac{X^2}{2} + C = \pm \frac{dY}{dX} \quad \dots \quad (\text{A-43})$$

where  $C$  is a constant of integration and is equal to zero, because when  $X$  equals zero,  $dY/dX$  equals zero.

Integrating Equation A-43 we have,

$$\left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) \frac{(a/b + 1)}{K} \frac{1}{D_o^2} \frac{X^3}{6} + C = \pm Y, \quad \dots \quad (\text{A-44})$$

where  $C$  is a constant of integration and is equal to zero, because when  $X$  equals zero,  $Y$  equals zero.

Expressing both  $Y$  and  $X$  in terms of diameters, Equation A-44 can be stated as

$$\frac{\Delta t_o \beta g D_o}{V_o^2} \frac{(a/b + 1)}{6K} \left(\frac{X}{D_o}\right)^3 = \pm Y/D_o \quad \dots \quad (\text{A-45})$$

When the  $N_{Pr}$  is taken as 0.7 and  $K$  equals 6.5, Equation A-45 becomes

$$0.065 \left(\frac{\Delta t_o \beta g D_o}{V_o^2}\right) (X/D_o)^3 = \pm Y/D_o \quad \dots \quad (\text{A-46})$$

Equation A-46 is an approximate trajectory equation from which the drop or rise of horizontally projected chilled or heated jets can be computed. Equation A-46 is compared with test data in the main body of this paper.

## DISCUSSION

H. B. NOTTAGE, Encino, Calif. (WRITTEN): Useful results, obtained by any reasonable means whatever, are always welcome additions to the engineering literature.

Having reached the present status, what does the author propose should be done regarding the unfilled fundamental needs?

S. F. GILMAN, Syracuse, N. Y., (WRITTEN): This is a very complex problem indeed and the author is to be congratulated for attacking it.

The slope of 3 in the approximation given by Equation 2 seems to be greater than that indicated by most of the data, the exception being Fig. 6. What needs to be pointed out is that Eq. 1, the exact equation, yields slopes increasingly above 3 as

$X/D_0$  increases. Referring to Fig. 5, the exact equation would start out along the line of Equation 2 ( $K = 6.5$ ) and then turn upward above the present curve. Therefore, it looks as if the approximate equation, Equation 2, yields better correlation than the exact equation, Equation 1, which should not be expected if the assumptions are reasonably valid. It may be that an important factor has not been considered in the development of the equations.

In any case, Equation 1, should be solved probably by numerical methods, for at least one condition, and compared with Equation 2. Certainly the restriction that the slope not be *too great* seriously limits the range of applicability of Equation 2. For example, since the term  $(dY/dX)^2$  in Equation 1 must be small in comparison with unity, a reasonable upper limit would be 5 percent of one, or 0.05. For this condition,  $dY/dX = 0.224$  which corresponds to an angle of 13 deg. Therefore, it can be expected that Equation 2 and Equation 1 will begin to deviate from each other quite markedly as the angle increases above 13 deg. The portion of the trajectory along which Equation 2 holds would then be only a small part near the nozzle (possibly out to the vertical white background line in Fig. 7). At an angle of say 45 deg, where  $(dY/dX)^2$  is equal to unity, Equation 2 would certainly not be valid. As previously stated, this looks like a case where the approximate equation provides a better fit than the exact. A considerable amount of additional analysis is evidently necessary, and it is hoped that Professor Koestel will pursue this problem further. He has done his usual fine job, but we still have a long way to go before we really understand the phenomena involved.

W. O. HUEBNER, Peekskill, N. Y.: This is another fine paper in the series of papers on air distribution, resulting from basic research at Case Institute of Technology. It may be of interest to note here that these papers are not only well-known in the United States, but that they are widely quoted in the literature of foreign countries such as Switzerland, the Scandinavian Countries, Germany, and others.

The new formula for the drop in height of horizontally projected chilled or heated jets developed by Professor Koestel is of immediate practical value in evaluating the performance of side wall outlets. To make it available for engineering use it is recommended to incorporate it into THE GUIDE at an early date.

As Chairman of the TAC Committee on Air Distribution, I would like to add a few words on some other work which we may expect from Case Institute.

We will have two more papers from Case Institute in the near future. The first of these papers deals with the performance and evaluation of room air distribution systems, which is not only a subject of practical interest, but also quite controversial. The other paper deals also with the very practical problem, namely, the performance of two parallel ventilating jets. Of course, we would like to know what happens if we put two air outlets too closely together. Then there is a third paper in preparation on radial outlets, which will be presented in the near future.

Professor Tuve has suggested—and the members of the TAC wholeheartedly agree—that it would be valuable to have in a single research bulletin the coordinated results of the research at Case Institute of Technology, dealing with air flow patterns from round, rectangular and slot type outlets, perforated panels, radial outlets, circular diffusers and typical residential installations.

I am convinced that you will concur with me if I ask Professor Koestel seriously to consider the preparation of such a bulletin.

**AUTHOR'S CLOSURE:** In answer to Dr. Nottage's question as to what should be done regarding the unfilled fundamental needs, I can only say that what we have been doing is not fundamental research but *problem solving* or analysis. We take what is made available from fundamental research and formulate useful engineering equations.

I am not sure what should be done as far as fundamental research is concerned, but I do know that we still have work to do in the laboratory developing more practical empirical equations for use in air distribution problems. Prof. Linn Helander of Kan-

sas State College is doing an outstanding job in the laboratory along this line. We are still in a position where we rely on basic researchers for our fundamental needs.

Dr. Gilman has made an excellent contribution to the discussion of our paper. It takes a lot of time and effort to dig into the mathematical details of this paper. The approximate Equation 2 has at least revealed the variables that should be considered when correlating test data. This is the important point. I am afraid that further mathematical analysis will only satisfy academic curiosity. However, I may be wrong so if time permits additional analyses will be made. More progress can probably be made if the general Equation A-7,  $Y/D_0 = f(X/D_0, N_R, N_{Frt}, N_F)$  appearing in the Appendix of this paper is evaluated experimentally. So far the mathematical analysis has not revealed the effect of the Reynolds number on the drop of the jet.

I am indebted to Mr. Huebner for his contribution to the discussion of this paper.

#### NOMENCLATURE

- $a$  = shape factor for the velocity profile.  
 $b$  = shape factor for the temperature difference profile.  
 $C_p$  = constant pressure specific heat, Btu per (pound) (Fahrenheit degree).  
 $C$  = constant of integration.  
 $C_1$  = constant of integration.  
 $C_2$  = constant of integration.  
 $D_0$  = nozzle diameter, feet.  
 $F$  = buoyant force at any distance  $X$ , pounds.  
 $g$  = acceleration due to gravity, feet per (second) (second).  
 $K$  = distance from nozzle face to the termination of the apparent constant velocity zone in diameters.  $K$  is the familiar constant of proportionality for air jets<sup>4</sup>.  
 $K_p$  = distance from apparent point source to the termination of the apparent constant velocity zone, in diameters.  
 $M_g$  = total jet momentum tangent to the centerline velocity axis at distance  $X$  from the nozzle, pounds.  
 $M_X$  = horizontal component of the total jet momentum,  $M_g$ , at distance  $X$ , pounds.  
 $M_Y$  = vertical component of the total jet momentum,  $M_g$ , at distance  $X$ , pounds.  
 $N_{Frt}$  = overall effective turbulent Prandtl number of the non-isothermal jet.  
 $N_F$  = ratio of the square root of the momentum force to buoyant force of gravity force and equal to  $\sqrt{V_0^2/\beta\Delta t_{0g}D_0}$  for a gas (also equal to the Froude number in hydraulics).  
 $N_R$  = Reynolds number.  
 $Q_0$  = heat flow or enthalpy flow of the jet at the nozzle measured above or below the ambient temperature, Btu per second.  
 $Q_s$  = heat flow or enthalpy flow of the jet in a direction tangent to the centerline velocity axis at distance  $X$  from the nozzle, Btu per second.  
 $r$  = radial distance from centerline axis, feet.  
 $r_1$  = radial distance from centerline axis to where  $V = V_1$ , feet.  
 $S$  = distance measured along the curved trajectory of a horizontally projected non-isothermal jet, feet.  
 $T_a$  = absolute temperature of the ambient air in degrees Rankine.  
 $V$  = air velocity at  $r$ , feet per second.  
 $V_1$  = air velocity at  $r_1$ , feet per second.  
 $V_0$  = maximum centerline air velocity at distance  $X$ , feet per second.  
 $V_o$  = air velocity at the discharge outlet, feet per second.  
 $X$  = horizontal distance from the apparent point source (assumed to be located on the nozzle face) based on the velocity profile, feet.  
 $Y$  = vertical displacement of the maximum velocity axis, feet.  
 $\beta$  = coefficient of expansion. For a perfect gas  $\beta$  equals  $1/T_a$  where  $T_a$  is the absolute temperature of the ambient air in degrees Rankine.  
 $\gamma$  = density, pounds per cubic foot.  
 $\Delta t$  = air-temperature difference in jet at  $r$ , Fahrenheit degrees.  
 $\Delta t_0$  = maximum centerline air-temperature difference at distance,  $X$ , Fahrenheit degrees.  
 $\Delta t_o$  = air temperature difference at the discharge outlet, Fahrenheit degrees.  
 $\rho$  = density, slugs per cubic foot.  
 $\theta$  = angular slope of the maximum velocity axis in degrees (see Fig. A-1).

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**1535**

## AIR CONDITIONING OF MULTI-ROOM BUILDINGS

By R. W. WATERFILL\*, NEW YORK, N. Y.

**T**HERE ARE several basic air conditioning systems and many modifications of each are presently applied to multi-room buildings. Principal ones are the single-duct primary with room secondary, the dual-duct, conventional low pressure central station, and room-unit systems. The single-duct primary and dual-duct systems will be compared in detail here and the other systems will be discussed generally.

### SINGLE-DUCT PRIMARY SYSTEM

Fig. 1 illustrates the single-duct primary system. It takes outdoor air, or a mixture of outdoor air with a small quantity of return air, which is filtered, cooled, and delivered in fixed proportions to each room unit. While it is possible to use only ventilation air in the primary with no recirculation return to the central apparatus, this is desirable only if ventilation requirements are relatively large.

Air is delivered at the room units under sufficient pressure to induce room-air recirculation flow through the room-unit coils, where it may be heated or cooled. The ratio of room-air circulation to primary air supplied may be 3 to 1 in volume and may range from 1 to 1, to 2 to 1 in room sensible heat capacity, being governed by noise levels, secondary condensation, air changes, and similar variables. The room coils are supplied with warm or cold water according to the operating needs of a particular zone, which accounts for the careful zoning required. At the peak of the cooling or heating season, the primary and secondary functions are the same, but at intermediate seasons the primary system may tend to overcool some rooms so that secondary reheat must be available in the room coils. Both air and water should be carefully zoned to have the necessary flexibility in each space conditioned.

### DUAL-DUCT SYSTEMS

Fig. 2 outlines the basic air handling and distributing principle of the dual-duct one-fan system. The entire air supply is filtered and delivered to warm and cold

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air distributing ducts. The flexibility of the warm and cold air dual supply insures individual room temperature control rather than zone control at all seasons of the year.

Among the principal advantages of the dual-duct system are its flexibility and automatic response to shifts in demands, without the necessity of cycle change-over as required by systems using warm or cold air or water alternately in a single-duct or water-pipe distributing network. Mixing units in each room draw air from both the warm and cold ducts in any desired proportion to satisfy the individual room temperature. Since there are two ducts, there is some tendency for the air volume at each outlet to vary. While this in no way affects temperature control, it could disturb distribution and air movement. Volume is, there-

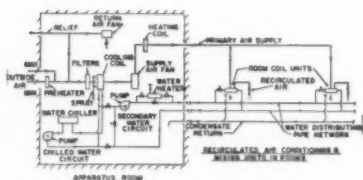


FIG. 1. SINGLE-DUCT PRIMARY SYSTEM

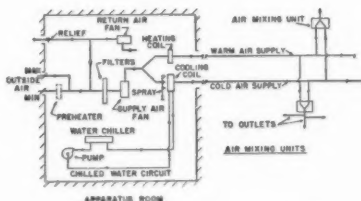


FIG. 2. DUAL-DUCT ONE-FAN SYSTEM

fore, maintained uniform by one of two means: (1) in zones by static pressure regulators, or (2) by volume regulators at each outlet.

Fig. 3 shows another dual-duct system, but with two fans arranged so as to effect a practical separation of return and outside air. Otherwise, its operation is the same as the dual-duct one-fan system. It is slightly more expensive and requires more apparatus floor space but it reduces refrigeration and reheat operating costs and maintains stable humidity over a wide range of conditions without reheat. Its advantages increase as the latent heat loads increase.

In the cooling season the warm air supply of the dual-duct systems is maintained at about room temperature, so that reheat is not ordinarily required, but it could be obtained in the separate warm-air duct by economical reverse cycle means, that is, from the refrigerant condenser. In the heating season, cooling and dehumidifying are obtained from outdoor air. This system is flexible and stable so that even a room with no internal heat load can achieve good control from the heat available from loaded rooms on the system. Heat generated in one area is automatically transferred to another heat-deficient room to produce good control without the addition of external heat from an artificial source.

#### CONVENTIONAL LOW-PRESSURE SYSTEM

The conventional low-pressure system has long been used where air conditioning load patterns are relatively stable and where recirculated air can be returned to the central conditioning apparatus for treatment or by-passed as zone conditions require. Control is accomplished at the central apparatus, thus is not flexible enough for multi-room applications.

## HIGH-PRESSURE CENTRAL SYSTEM

Fig. 4 shows a high-pressure central system which uses reduced air volumes at high temperature differentials, and relatively high duct velocities. Its single supply temperature for each zone is established in the central plant; thus variations in individual rooms must be compensated for by throttling the air supply volume. Reductions of more than 25 percent upset distribution and circulation. When 100 percent outside air is supplied to the enclosure, heat and refrigeration requirements are excessive and adequate relief or exhaust means must be provided. Windows are unsatisfactory reliefs since they admit noises, dirt, and wind and are unpredictable. For this and other reasons, this system is not included in the detailed analysis.

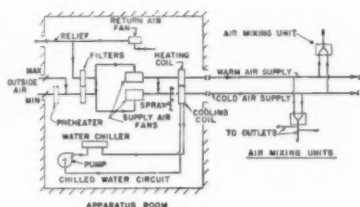


FIG. 3. DUAL-DUCT TWO-FAN SYSTEM

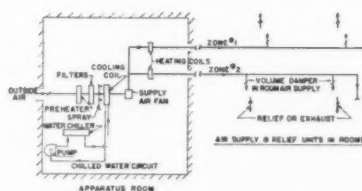


FIG. 4. MODIFIED CENTRAL STATION SYSTEM WITH 100 PERCENT OUTSIDE AIR—NO RECIRCULATION

## ROOM-UNIT SYSTEMS

Room-unit systems are of dubious adaptability to multi-room buildings, chiefly because of their control and ventilation characteristics. They are either filter fan and coil units which are supplied with chilled or hot water from a central plant, or self-contained units which include a refrigerating system. The electric wiring, water, ventilation, filtering, humidity characteristics and servicing of a vast number of such units generally precludes their consideration in large buildings. Servicing problems tend to multiply with time. For interior zones, ventilation and condenser problems increase. The partial load performance of room units is also poor, resulting in a high seasonal average power consumption.

## VENTILATION AND FILTERING

Ventilation is a vital function of air conditioning. Present air conditioning practice generally provides more positive and generous ventilation than was ever enjoyed in non-air-conditioned buildings and possibly even in some sections of open-air sports stadiums.

Good engineering demands that the thermal and circulation requirements be considered independently of ventilation. The proper design approach is to determine ventilation requirements from the nature of the occupancy and accepted codes or practices of recognized groups. Excessive or fixed ventilation can be wasteful.

Fig. 5 gives some indication of the cost of outdoor air at temperature extremes. For instance, a cooling system using 100 percent outside air at 75 F wet bulb requires a refrigeration plant capacity considerably in excess of a system restricted to ventilation needs at these peak conditions. Assuming for illustration an average internal sensible heat load of 25 Btu per (hr) (sq ft) of floor area and a 50 F cold-air supply temperature, the calculated air volume required to maintain an 80 F room temperature is 0.80 cfm per sq ft. Fig. 5 shows that if this total volume is taken from outdoors, the refrigeration demand due to the ventilation air would

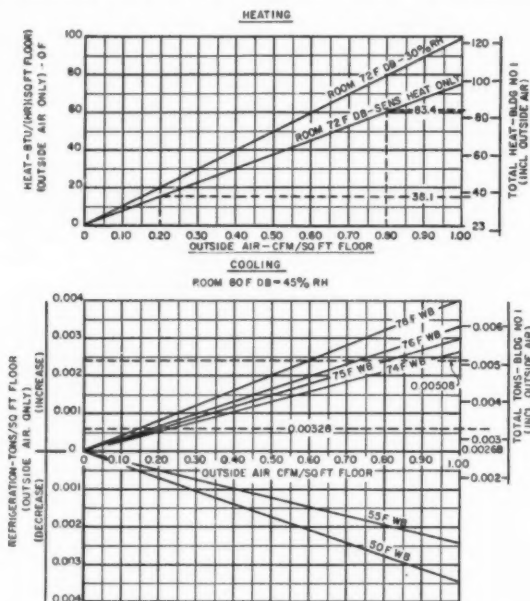


FIG. 5. VENTILATION LOAD PER SQUARE FOOT OF FLOOR AREA

be 0.0024 tons per sq ft of floor area for the example illustrated, whereas if the outdoor air is limited to the ventilation requirement of 0.20 cfm per sq ft, the re-  
frigeration demand due to the ventilation air would be 0.0006 tons per sq ft. The  
corresponding *total* load would be 0.00508 and 0.00328 tons respectively. This is  
not an extreme case, yet the use of 100 percent outdoor air would require a re-  
frigeration plant 1.55 times as large as need be.

When outdoor air wet-bulb temperatures are lower than interior wet-bulb temperatures, it is decidedly advantageous to make use of available outdoor air to reduce the cooling load and to reduce the required hours of operation of refrigeration equipment, as shown in Fig. 5.

For example, if the outside air is limited to the minimum ventilation value of 0.20 cfm per sq ft at 55 F wet bulb, its cooling capacity in an 80 F room is only 0.0005 tons. This still leaves approximately 0.0015 tons per sq ft mechanical

refrigeration requirement, allowing a transmission load reversal of 0.0007 tons. However, utilizing the full system capacity of 0.80 cfm outside air per sq ft or four times the minimum, provides 0.002 tons cooling capacity and permits shutting down the mechanical refrigeration system.

During the heating season, the penalty of 100 percent outdoor air is even greater, the required capacity of the heating plant being nearly double that of a system which provides for recirculation. There are special situations like certain hospital wards where recirculation between zones is not permissible, but this does not alter good practice in general multi-room structures.

The purpose of ventilation is the dilution of impurities. Present tendencies are toward overemphasis, except in public enclosures like subways. Window area is often used as a ventilation index, though windows in non-air-conditioned structures are often closed, weather-stripped and sealed. Offices on the leeward side of such buildings get no direct ventilation, the windows serving merely as air relief vents for the rest of the building. Windows in non-air-conditioned structures are generally opened only for the available cooling effect of outside air and not for true ventilation purposes.

Ventilation should be based on some fixed criterion such as cubic volume or square feet of floor area of the conditioned space or on the population density, and should take into account the activity in the building. Industrial fumes and food odors require special treatment not considered here. Basing ventilation on percentage of fan capacity is erratic. Current ventilation standards, when adhered to, result in a high purity of the air in conditioned spaces.

Another desirable function of air conditioning, rivaling ventilation, is physical cleanliness of the air. Conditioned air in cities should be, and generally is, cleaner than outdoor air. Some dirt originates within the conditioned space itself and is being constantly raised and resettled. A desirable feature of returning recirculated air to a central apparatus is that a large volume of the room air and dust can be filtered in efficient types of filters. The maintenance of a central filter system is more effective and the cost of air filtration is lower than for similar usage in a dispersed filter system. The cleaning of room units, if not equipped with efficient individual filters, is an added problem and is comparatively costly.

The room filtered air change rate is important in maintaining high standards of cleanliness. When the air is recirculated through widely scattered room conditioning coils, dirt and odors tend to accumulate, with the result that ventilation objectives are largely lost.

With the single-duct primary and dual-duct one-fan systems, the ventilation to all conditioned spaces is effective under any control position, that is, no space will receive less dilution of its air volume by outside air than established by the system adjustment. Ventilation in a dual-duct two-fan system under some conditions could be reduced, but probably not more than 10 percent in any room, if the warm air is maintained high enough to balance the excess cooling effect of the ventilation air in winter.

With the single-duct primary system, use of outdoor air and filtering is limited generally to the primary air supply. The dual-duct system can supply up to 100 percent ventilation and does filter the total air quantity at the central apparatus.

#### HUMIDITY

Room humidity depends on the internal moisture load, the moisture content of the air supplied to the room and the room sensible heat or reheat. Close control



difficulty is that the circuit may have to be changed over from reheat to cooling and back again in a few hours.

The humidity behavior of the dual-duct one-fan system is affected by the outside air moisture content. Figs. 8 and 9 are presented to show this effect. Line *A* shows the change in zone humidity with variations of room sensible heat. Reheat applied when this line crosses the 50 percent relative humidity line holds room conditions along *A'*. It will be noted that in both Figs. 8 and 9, reheat is not

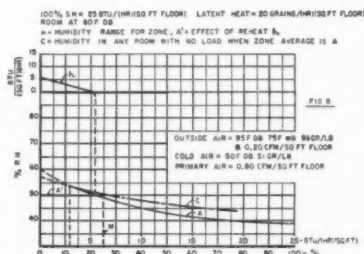


FIG. 8. HUMIDITY BEHAVIOR OF DUAL-DUCT ONE-FAN SYSTEM WITH 98 GRAINS PER LB OUTSIDE AIR

needed when internal sensible heat exceeds 20 and 25 percent. Even the individual room with no internal load, indicated by line *C*, maintains a temperature of 80 F and a relative humidity below 50 percent when it is on a zone supply having an average sensible heat in excess of 30 percent. This is because the dual-duct system utilizes the excess heat of loaded spaces to reheat unloaded areas.

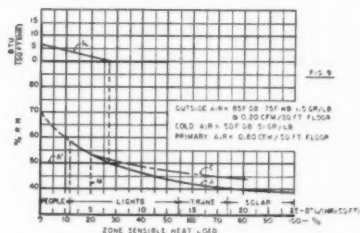


FIG. 9. HUMIDITY BEHAVIOR OF SAME SYSTEM WITH 115 GRAINS PER LB OF OUTSIDE AIR

The amount of applied reheat required is the internal sensible heat deficiency, where the humidity begins to exceed its desired value along line *A*. Thus, in order to prevent the humidity from exceeding 50 percent and make it follow line *A'*, reheat begins where *A* crosses the 50 percent relative humidity line and must be applied in the amount *h*, in Btu per square foot of floor area per hr, as shown

in the upper scale of the figures. While reheat at these outside air temperatures is unlikely, it is well to have a small amount available at lower temperatures in the interest of good control. The dual-duct system with its separate warm air duct and heating coil in the apparatus room is ideally arranged to obtain this interim reheat from the condenser side of the refrigerant cycle. The humidity curve for the dual-duct two-fan system is flatter than that of the corresponding dual-duct one-fan system.

### TYPICAL APPLICATION

To illustrate the behavior of systems currently employed in multiroom buildings, a representative New York office building, designated Building 1 (Table 1), with its air conditioning loads is assumed. These loads are average over the total floor area and are instantaneous peaks, instead of totals of all zone peaks, and will vary from zone to zone and from time to time. The distribution system must have the capacity and flexibility to meet all zone peaks and load variations. These comparisons will apply either to an entire building or to any segment of a building, including the periphery, the load averages being as assumed. Building 1 is analyzed in detail, but to illustrate the effect of heavier thermal loading, condensed data on Building 2 are given. In addition to assumed conditions, Table 1 summarizes unit results for both buildings, employing single-duct primary, dual-duct one-fan, and dual-duct two-fan air distribution systems.

TABLE 1—HEATING AND COOLING LOADS, PRINCIPAL AIR QUANTITIES AND UNIT ANNUAL OPERATING COSTS OF A TYPICAL MULTI-ROOM BUILDING IN NEW YORK

1. COOLING & HEATING LOADS — BTU/HR/SQ FT (AVERAGE OF FLOOR AREA)									
2. SUMMER 55°F DB 75°F WB OUTSIDE AIR & 80°F DB ROOM; WINTER 0°F DB OUTSIDE AIR & 72°F DB ROOM									
3. BUILDING									
	NO. 1				NO. 2				
	COOLING	HEATING	COOLING	HEATING	COOLING	HEATING	COOLING	HEATING	
	BTU	KWH/HR	BTU	KWH/HR	BTU	KWH/HR	BTU	KWH/HR	
4. TRANSMISSION	4.9	19.2	23	6.6	18.9	5.3	31.7	9.1	
5. SOLAR	6.6	26.4			13.0	37.1			
6. LIGHTS (3 WATTS/SQ FT BUILDING NO. 1) (3 1/2 WATTS/SQ FT BUILDING NO. 2)	10.7	42.8			12.5	35.7			
7. PEOPLE — SENSIBLE HEAT (1 1/2 Btu/100 sq ft)	2.9	11.5			2.9	8.3			
8. ROOM — SENSIBLE HEAT TOTAL	23.0	100.0			35.0	100.0			
9. PEOPLE — LATENT HEAT	3.1				3.1				
10. FAN & PUMP	4.0				5.1				
11. VENTILATION (0.20 CFM/SQ FT + 16 CFM/PERSON)	7.3		15.1		7.3		15.1		
12. INFILTRATION (0.10 CFM/SQ FT)			7.55				7.55		
13. TOTAL	39.4		46.15		50.5		54.75		
14. TONS/SQ FT		.00328				.0048			
15. FLOOR AREA/1000 TONS — SQ FT		305,000				236,000			
16. COLD AIR SUPPLY TEMP. — °F									
17. PRIMARY AIR CHANGES/HR	8.5	5.3	5.3	3.33	7.4	7.4			
18. PRIMARY CFM/SQ FT	.42	.80	.80	.50	1.12	1.12			
19. ROOM CIRCULATION — CHANGES/HR	11.0	11.0	11.0	13.5	15.0	15.0			
20. SENSIBLE HEAT CAPACITY PRIMARY-DOFPM	11	25	25	13	35	35			
21. ANNUAL OPERATING COST/SQ FT	\$0.22	\$0.202	\$0.194	\$0.288	\$0.269	\$0.258			
22. ANNUAL OPERATING COST/TON CAPACITY	\$67.2	\$61.7	\$59.3	\$98.5	\$64.2	\$61.5			
23. VENTILATION DURING WORK HOURS ONLY; INFILTRATION — IDLE HOURS									
24. IF 55° SUPPLY IS USED THE REFRIG. KW/HR REDUCTION APPROX FAN KW/HR INCREASE									
25. X THESE AIR VOLUMES ARE IN HEAT BALANCE WITH THE PEAK BUILDING LOADS AND REFRIGERATION PLANT THE FAN CAPACITY IN EACH CASE MUST EXCEED THESE VALUES BECAUSE OF SOLAR AND INTERNAL LOAD SHIFTS									

Ventilation is 0.20 cfm per sq ft of floor area, equal to 16 cfm per person, and is used only when the building is occupied. During nights or holidays ventilation

becomes infiltration of 0.10 cfm per sq ft, or equivalent to one air change every 90 min in the building. The refrigeration load at design conditions for the two buildings becomes 0.00328 tons and 0.0042 tons per sq ft. A refrigeration plant of 1000 tons capacity is then adequate for buildings having 305,000 and 238,000 sq ft, respectively.

The cold air supply temperatures and primary air volumes were selected as representing reasonable and preferred practice, although they can be varied within limits. The cold air dry-bulb temperatures result from choosing a common apparatus dew point, which is established ahead of the fan of the single-duct primary draw-through system and after the fan of the blow through dual-duct system. The difference is approximately the rise after the fan of the single-duct primary system.

If the cold air supply temperature of the dual-duct systems is increased to 55 F, the primary air supply is also increased, but the refrigeration load and hours of operation of the refrigerating system are decreased, with the result that the combined fan and refrigeration kilowatt hours remain substantially unchanged. It is also possible to reduce the primary air volume of the single-duct primary system, but this would again extend the required operating time of the refrigerating system, as well as reduce primary air change in the room. Thus, slight changes in these assumed values would have only slight effect on the final results.

The primary air volumes given in Table 1 are based on the combination of zone loads which give a maximum load for the building and correspond to the refrigeration plant capacity. Due to solar and internal load shifts such as are experienced in cafeterias, conference rooms and similar areas, the total of all zone peak loads exceeds the instantaneous total. Since the air volumes to each zone are fixed, they must be adequate for all zone peaks, that is, they must be in heat balance with all zone peaks, and therefore, the fan capacity of all systems must exceed these values. These volumes, therefore, have been increased 10 percent in estimating the kilowatt-input of the supply and return air fans. The primary air volumes also represent the filtered air supply to the rooms.

The annual operating cost for electric power, steam and water per sq ft of floor area for the two buildings is almost proportional to their thermal loadings or approximately the same on a refrigeration tonnage basis.

#### REFRIGERATION AND HEAT DEMAND

Operating costs based on peak load conditions are apt to be misleading unless mean load factors for the specific combinations are well known. This is particularly true in determining refrigeration power input, because of the reduced efficiency of refrigeration plants at light loads, and because some systems operate at reduced loads longer hours than other systems. In order, therefore, to establish the demand for refrigeration power and for heat in relation to changes in outdoor temperatures, the curves, Figs. 10, 11 and 12, were developed. Unit systems would have a much higher average power consumption. These curves include the progressive changes in heat transmission through walls, ventilation loads and use of extra outside air where advantageous. When the duration of outside temperatures for the various hours of the day, throughout the year, is found from New York City weather records, and the increments of load for each temperature are known, the cumulative power and heat requirements can be determined quite accurately.

The refrigeration load is based on the hours indicated and on the outside air

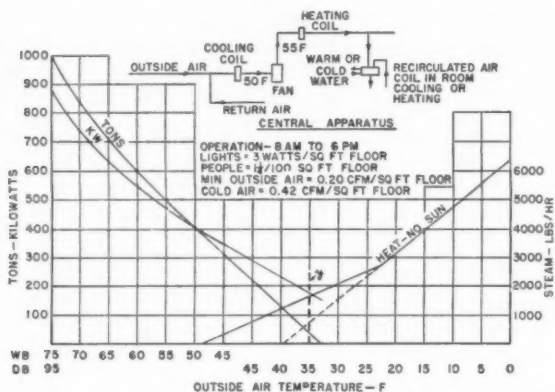


FIG. 10. SINGLE-DUCT PRIMARY SYSTEM—OFFICE BUILDING WITH 1000 TONS—305,000 SQ FT FLOOR AREA = 0.00328 TONS PER SQ FT OF FLOOR

wet- and dry-bulb temperature records. The kilowatt-input is based on a single centrifugal refrigeration compressor using a cooling tower. The refrigeration tonnages include full sun heat, but allowance has been made for cloudy and short days when outside temperatures are below 60 F wet bulb. The temperature at  $y$  indicates the change-over point from refrigeration to heat for a primary cold air supply temperature of 55 F and line  $x$  indicates the same for a supply temperature of 50 F. When the sun is not shining, the changeover point  $y$  on the single-duct primary system can be moved to 42 F outside air dry-bulb temperature, and this is considered in determining operating hours.

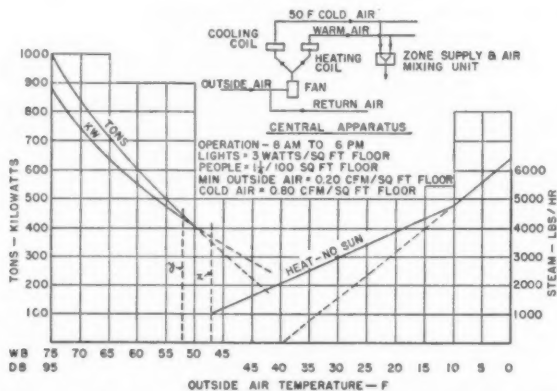


FIG. 11. DUAL-DUCT ONE-FAN SYSTEM—OFFICE BUILDING WITH SAME LOAD AS FIG. 10

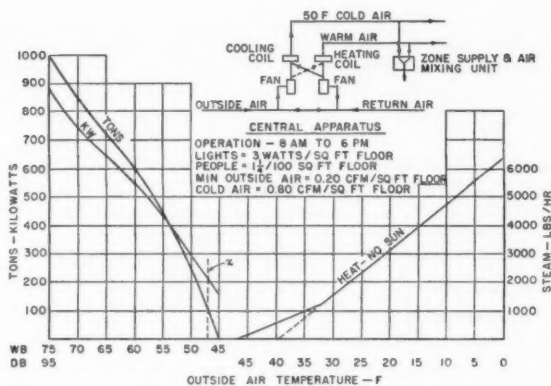


FIG. 12. DUAL-DUCT TWO-FAN SYSTEM—OFFICE BUILDING WITH SAME LOAD AS FIG. 10

The daytime heat requirements are based on outside dry-bulb temperature with no assistance from the sun, and the base scale is so arranged as to indicate the progressive transition from the cooling to the heating cycle. The diagonal line with the broken line extension is the heating load with the minimum ventilation specified, whereas the solid line extension is the actual heat used for additional outside air employed to permit the earliest possible shutdown of the refrigeration plant. The continuation of the refrigeration or heat demand lines beyond the change-over point merely shows the trend of these curves and is not included in the calculations.

For the dual-duct one-fan system (Fig. 11) the outside air volume which is large at  $x$  could be reduced so that refrigeration could continue along its extension load conditions, but in practice a small amount of reheat must be available for room variations and for heat load shifts for any system. It may be noted that with the dual-duct two-fan system there is an interval when neither refrigeration nor heat is indicated.

For winter nights and holidays, or idle hours, it is assumed that ventilation will be reduced to actual infiltration and that a night temperature of 65 F is maintained. The heat requirements at these periods are the same for each of these systems.

The distribution of heat in an unoccupied building varies, but it is recognized that full operation of the air handling system is unnecessary and undesirable with any system. Reduced operation can be achieved in most winter weather conditions by reducing primary fan speed or by operation of the return air fan only in dual-duct systems and by hot water circulation in the single-duct primary or in any system with room secondary conditioning units or radiators.

#### OPERATING COSTS

The analysis of annual operating cost, Table 2, is based on an installation requiring a 1000-ton refrigeration plant, for convenience and because refrigeration

capacity is a fairly good index for similar systems in different buildings. A central station or any other system, however, using 100 percent outside air at temperature extremes, would require a refrigeration plant of 1550 tons and a heating plant capacity about twice that of the systems analyzed in Table 2 for Building 1, even if the air volume is throttled 25 percent in winter.

TABLE 2—ANALYSIS OF TOTAL ANNUAL COSTS FOR A TYPICAL MULTI-ROOM BUILDING IN NEW YORK

1. AIR CONDITIONING SYSTEMS—OPERATING POWER & WATER COSTS.			
2. INTERNAL SENSIBLE HEAT LOAD = 25 BTU/HR/SQ FT AVG.; VENTILATION = 0.20 CFM/SQ FT.			
3. AIR DISTRIBUTION SYSTEM	SINGLE DUCT PRIMARY	DUAL DUCT	
		1—FAN	2—FAN
4. TOTAL HOURS OPERATION—REFRIG. & C.T.	1726	1137	1137
5. OPERATING KW. HRS. REFRIG. (1000 T.—880 KW)	937,000	682,000	655,000
6. OPERATING KW. HRS.—FANS & PUMPS (2600 HRS.)	1,054,800	1,134,000	1,134,000
7. OPERATING KW. HRS.—COOLING TOWER (FAN & PUMP)	219,000	147,000	147,000
8. TOTAL KW HRS. (WORK HRS.)	2,210,800	1,963,000	1,936,000
9. POWER COST AT 2¢/KW. HR. (2600 HRS.)	\$44,210	\$39,260	\$38,720
10. STEAM COST AT \$1.00/1000 LBS.	\$ 2,200	\$ 2,800	\$ 1,050
11. COOLING TOWER MAKE-UP WATER AT 25¢/1000 GAL.	\$ 1,170	\$ 860	\$ 800
12. TOTAL—POWER, STEAM, WATER (2600 HRS.)	\$47,580	\$42,920	\$40,570
13. OPERATING KW HRS. FAN OR PUMP (IDLE HRS.)	327,000	283,800	283,800
14. POWER COST AT 2¢/KW. HR.	\$ 6,540	\$ 5,676	\$ 5,676
15. STEAM COST AT \$1.00/1000 LBS.	\$13,100	\$13,100	\$13,100
16. TOTAL COST—POWER, STEAM (IDLE HRS.)	\$19,640	\$18,776	\$18,776
17. TOTAL ANNUAL COST—POWER, STEAM, WATER	\$67,220	\$61,696	\$59,346

The total hours of operation of the refrigerating equipment emphasizes the advantage of flexible and free use of outside air and the large primary air supply capacity of the dual-duct systems. The operating costs of power are based on a uniform 2 cents per kwhr rate, whereas the longer operating period of the single-duct primary system would increase the demand charge influence and raise the mean rate in the extended period of low refrigeration usage. This would increase the power cost of the single-duct primary system slightly.

The air conditioning systems are considered to operate fully 10 hr daily for 260 days. When the building is not occupied, operation is reduced to the minimum heating needs of transmission and infiltration losses. Heat is distributed by the return air fan of the dual-duct systems and by the hot water pumps of the single-duct primary system.

The occupied and unoccupied periods are separated and subdivided, so that costs for different conditions can be estimated. If steam is used for a turbine-driven centrifugal refrigerating unit, a steam rate of 16 lb per hp-hr and \$1.00 per 1000 lb would equal the cost of the motor driven unit at 2 cents per kwhr. Any steam system, however, would increase the size and operating cost of the cooling tower.

Static pressures on the supply fans are assumed to be 8 in. for the single-duct primary and 6 in. for the dual-duct systems. All return fans are figures for 2 in. static and all pumps for 100 ft head. These values are believed representative.

The total annual power, steam and water costs show the dual-duct two-fan system to be lowest, with a maximum difference of under 10 percent. The unit values are indicated in Table 1, Lines 21 and 22.

To the annual operating costs of Tables 1 and 2 must be added other charges. The initial investment cost of air conditioning equipment alone, exclusive of building changes, foundations, electric power, steam, water and sewer facilities, is about \$800 to \$1200 per ton of refrigeration capacity, with the dual-duct one-fan system generally lower than the single-duct primary system when designed specifically for the respective systems. The fixed charges on this investment at 6 percent interest and 15-year depreciation with no salvage value equals 10 percent or \$80 to \$120 per ton annually.

The single-duct primary system duct and piping system when used on the building periphery and in combination with a central station system for the interior zones will occupy about 2.5 percent of the building floor area. The dual-duct one-fan system when applied to both periphery and interior zones will require about 3 percent of the floor area for its installation. The average value of this space, including basement, is about \$3.00 per sq ft. Maintenance and operating supervision for the single-duct primary system is estimated at about 25 percent higher than for the dual-duct system to allow for additional cleaning of the coils and boxes of the 500 to 600 room units, for servicing water valves and for shifting the secondary water from heating to cooling to meet seasonal and daily midseasonal load changes. The total annual fixed charges are about 2.5 times the power, steam and water costs.

## APPENDIX

Determination of relative humidity in rooms conditioned by dual-duct one-fan system:

$$G_x = [G_c (1 - P_w) + G_o P_w P_o + G_r] \div [1 - P_w (1 - P_o)]$$

$$G_w = G_x + P_o (G_o - G_x)$$

$$P_w = (t_x - t_c) / (t_w - t_c), \text{ for sensible heat balance}$$

$$P_w = (G_x - G_c - G_r) \div [G_x - G_c + P_o (G_o - G_x)], \text{ for grains moisture balance}$$

$$t_w = [t_o - t_c (1 - P_w)] / P_w$$

$G_x$  = moisture in return air, grains per pound.

$G_c$  = moisture in cold air supply, grains per pound.

$G_o$  = moisture in outside air entering mixing chamber, grains per pound.

$G_w$  = moisture in warm air supply, grains per pound.

$G_r$  = moisture load in room, grains per pound of supply air.

$P_w$  = warm air, percent of supply.

$P_o$  = outside air, percent of supply.

$t_o$  = cold air supply temperature.

$t_w$  = warm air supply temperature.

$t_x$  = room supply temperature.

## DISCUSSION

JOHN S. KING, Fairfax, Va., (WRITTEN): The author has prepared a very interesting study of some of the fundamental concepts required for the rational design of air conditioning of multi-room buildings. The author's statements comparing the possibilities and behavior of the many variations of duct systems are evidently based upon many years of experience.

The nine graphs presented are of interest not only because of content, but also be-

cause they demonstrate the clarity of graphical representation of the many factors contributing to good air conditioning design.

The author's presentation of estimated operating costs based upon the graphical portrayal of operating (demand) loads changing with the rise or fall of outdoor air temperatures, together with the influence of three different systems of ducting, are an interesting study which has been incorporated into Table 2, which shows the total (power, steam, and water) annual operating costs based upon the three indicated systems of ducting.

In regard to the latter (Table 2) the writer would like to see presented therein the estimated construction (installation) costs and finance charges for the three duct systems compared (and also the estimated construction costs and finance charges for the balance of the air conditioning installation), as components of the important over-all economy.

The writer would like to hear the author comment more fully upon the characteristics of the dual-duct system (which is presented in Fig. 3), and which is described at some length.

The author's final descriptive statement: "This system is flexible and stable, so that even a room with no internal heat load can achieve good control from the heat available from the loaded rooms on the systems. Heat generated in one area is automatically transferred to another heat-deficient room to produce good control without the addition of external heat from an artificial source", is not clear to the writer.

ALBERT GIANNINI, New York, (WRITTEN): The author has done a substantial amount of work indicating considerable thinking on the subject. It is a broad and exceedingly important topic concerning which there is naturally some divergence of opinion and apparently, some conflict in experience. It is to be regretted that space limitations have forced the author to limit the scope of his paper at the expense of several important aspects of the subject.

In discussing the single-duct primary system, the paper states that there is a need for careful zoning of both air and water. The fact is that the single-duct primary system is not a zone control system, but by its very nature is an individual room control system. While modern practice may indicate the use of multiple sets of apparatus because of convenience, they can be and usually are operated as a single functional zone. For example, this is so on the largest post-war office building project, namely the Gateway Center in Pittsburgh where approximately 6000 units and 4500 tons of refrigeration are used.

This is possible because of a basic design characteristic of the single-duct primary system. Namely, that the necessary flexibility is obtained by scheduling reheat of the primary air as a function of the outside temperature. Thus the possibility of overcooling any room, as suggested by the author, does not exist, since the only portion of the total load which can become negative is the transmission load which also is a function of the outside temperature. All other loads encountered are positive. Each unit's capacity is matched to these loads by simple throttling with a water control valve at each unit thereby affording individual room control.

Without overcooling the adverse humidity conditions indicated in Fig. 6 of the paper never occur in any space. Therefore, Fig. 6 is misleading. Experience on over 400 single-duct primary air systems gives ample evidence that no humidity problem exists. There is no difficulty in consistently maintaining humidities below a design maximum.

In describing the humidity variation of the dual-duct one-fan system, (Fig. 8) it is stated that "in order to prevent the humidity from exceeding 50 percent and make it follow line A' reheat begins where A crosses the 50 percent relative humidity line. . . ." In practice, the room thermostat responds to a change in sensible load, and when the load decreases, a greater percentage of air from the hot duct is admitted. But this air is *not dehumidified* so that the room humidity must build up. In addition, the formulae given in the appendix, for calculating relative humidity in rooms conditioned by dual-duct one-fan systems, imply a steady state condition. In a room or group of

rooms lightly loaded due to shading or absence of lights for any period of time, there is bound to be a gradual and continuous rise in humidity.

Experience indicates that instability of the dual-duct system (which is a volume control arrangement) is dismissed in the paper too easily. Many people are cognizant of this as a real problem and it is interesting to note that in the same issue of *Heating, Piping & Air Conditioning*, in which this paper was published an article by C. Milton Wilson states: "Engineers are rightfully concerned over a static unbalance between the hot and cold ducts at a particular mixing valve. Unless proper precautions in both duct sizing and controls are taken, the unbalance would occur when a majority of either the hot or cold valves on a particular zone were closed by thermostatic action". This article points out that this unbalance can cause noise and that careful design is required. The suggestion is made that this unbalance can be minimized by a combination of temperature and static pressure control. As the flow in the hot duct decreases, the temperature of the hot duct is lowered with the objective of keeping the flow fairly constant in both hot and cold ducts.

Another possibility for maintaining stability is to vary the static pressure developed by the fan as the flow changes. This calls for instruments responsive to flow or velocity pressure. But in any case, system stability requires more than simple S.P. regulation.

The author points out that a two fan system is required to minimize the amount of undehumidified outside air entering the system. However, the outside air supplied to rooms will then vary with their sensible load. Therefore, in the interest of both humidity and ventilation control, hot air duct quantities should be kept low by raising the temperatures, as the author suggests. However, as Wilson points out, system stability calls for keeping hot air duct quantities up by lowering hot duct temperature. There appears to be a fundamental conflict here.

In the author's analyses of system performance of a typical office building, it is not clear whether the figures are confined to the periphery of the building, or if interior sections are included since mention is made of such areas as cafeterias and conference rooms; but since the single-duct primary system is applied only to exterior portions of the building, it must be assumed that only the periphery is included.

It is stated that slight changes in assumed values of primary air cfm and cold air supply temperatures would have only slight effect on the final results of the analysis. This deserves some examination. Using Building #2 figures, the cooling requirements of 35 Btu per sq ft can be easily met with commercially available induction units with 0.4 cfm per sq ft or lower of primary air. This is a 20 percent reduction over that indicated. If an additional 2 deg were allowed for heat gain in the supply duct 0.4 cfm per sq ft of primary air would still be sufficient with typical modern induction units to handle the total load, whereas this same assumption of 2 deg rise in the supply air would increase the cfm required by the double-duct system to 1.2 cfm per sq ft instead of 1.12 cfm per sq ft. Hence, it is entirely reasonable for the air required by the double-duct to be 300 percent of that required by the single-duct primary system rather than 225 percent indicated in the comparison. This is an important difference, not only because of the additional amount of air to be pumped, which requires larger ducts, fans and motors but also each of these ducts must be increased to handle the greater total quantity.

From a practical point of view, the indicated air requirements of a double-duct system seem low. There is a growing trend towards the use of a room design temperature lower than 80 F at peak loads, and at partial loads, 80 F is intolerable. For instance; if the outdoor prevailing temperature is 85 F instead of 95 F, a room temperature of 75 F or 76 F would have to be maintained. There is very little change in total load due to this small change in transmission load. For example, with a 75 deg room and 85 deg outside temperature, the btu per sq ft would be approximately 33 instead of 35. If a 2 deg rise in temperature is assumed in the duct from central plant to room, then the temperature of the air leaving the cooling coils would be as follows if the tabulated 1.12 cfm per sq ft is used:

$$\frac{33}{1.12 \times 1.08} = 27 \text{ deg temperature difference between room and supply air.}$$

$$27 + 2 = 29 \text{ deg difference between air leaving coils and room air temperature.}$$

$$75 - 29 = 46 \text{ deg air leaving coils.}$$

This is a rather low figure and would increase the horsepower per ton on the refrigeration machinery. If the suggested 50 F supply air temperature is used then:—

$$\text{cfm/sq. ft.} = \frac{33}{1.08 (75 - 50)} = 1.22$$

This is 10 percent more than the design quantity indicated in the example.

Of course, if a design room temperature lower than the 80 F were used at peak load, then the air requirements would be even greater and the amount of air to be handled by the dual-duct system would be even greater than 300% of that required by the single-duct primary system.

The basis for arriving at the difference in the operating hours assigned to the refrigeration plant for the two different systems is not given. It would be interesting to have this information because it is not an easy thing to calculate and requires innumerable assumptions. The number of hours of operation per day and the capacity required by the refrigeration plant not only varies with dry bulb but with wet bulb and particularly with sunshine. Figs. 10, 11, and 12 in the author's paper are, therefore, questionable without further amplification. A difference in the refrigeration hours in the order of 10 to 15 percent instead of the 33 percent indicated in the paper would seem more reasonable provided the *all air* system used fairly high supply air temperatures.

While it is very difficult to analyse refrigeration operation costs, there should be no doubt about the operating cost requirements for pumping the fluids required in any system. For purposes of comparison some accurate figures are given herewith.

#### FOR SINGLE-DUCT PRIMARY SYSTEM

$$\begin{aligned} 0.4 \text{ cfm per sq ft} &= \text{Total Air} \\ 0.2 \text{ cfm per sq ft} &= \text{Return Air (Total Air minus 0.2 cfm per sq. ft. outside air)} \\ \text{Power for primary air fans} &= 0.00180 \text{ HP/cfm} \\ \text{Power for return air fans} &= 0.00045 \text{ HP/cfm} \\ \text{Power for water circulating pump} &= 0.0007 \text{ HP/cfm of primary air} \end{aligned}$$

Primary air fans are figures at 8 in. static pressure and return air fans at 2 in. static pressure. The figure for HP/cfm for the pumps contemplates the usual ratio of gpm to cfm for an induction unit and estimates a 100 ft head on the pump. Fans are figured for 70 per cent static efficiency.

$$\text{Fan HP} = \frac{\text{cfm} \times \text{S.P.} \times 0.000157}{\text{Static EFF (70\%)}}$$

Then,

$$\begin{aligned} \text{Primary air HP} &= 0.4 \times 0.00180 = 0.000720 \text{ HP per sq ft.} \\ \text{Return air HP} &= 0.2 \times 0.00045 = 0.000090 \text{ HP per sq ft.} \\ \text{Water pump HP} &= 0.4 \times 0.0007 = 0.000280 \text{ HP per sq ft.} \\ \hline \text{Total} &= 0.00109 \text{ HP per sq ft.} \end{aligned}$$

#### FOR THE DUAL-DUCT SYSTEM

$$\begin{aligned} 1.2 \text{ cfm per sq ft} &= \text{Total Air} \\ 1.0 \text{ cfm per sq ft} &= \text{Return Air} \\ \text{Power for supply air fan (6 in. SP)} &= 0.00135 \text{ HP/cfm} \\ \text{Power for return air fan (2 in. SP)} &= 0.00045 \text{ HP/cfm} \\ 1.2 \times 0.00135 &= 0.00162 \\ 1.0 \times 0.00045 &= 0.00045 \\ \hline \text{Total} &= 0.00207 \text{ HP per sq ft.} \end{aligned}$$

The HP requirements for pumping the fluids on a single-duct primary system for the building used as an illustration by the author would be:  $238,000 \times 0.00109 = 260$  HP. Whereas for the dual-duct system, the figure would be:  $238,000 \times 0.00207 = 492$  HP.

If we assume 9/10 of a kw per HP and use the author's figure of 2 cents per kwhr and an operating period of 2600 hrs per year, then:

$$260 \text{ HP} \times 0.9 \times 0.02 \times 2600 = \$12,200 \text{ per yr. (For single-duct primary system)}$$

$$492 \text{ HP} \times 0.9 \times 0.02 \times 2600 = \$23,000 \text{ per yr. (For dual-duct system)}$$

This means there is about \$10,800 a year extra operating cost for pumping fluids on the dual-duct system for this particular building. These figures seem incontrovertible to the writer. This appears to be a large penalty to pay for some possible refrigeration operating cost saving. Even if one accepts the author's tabulated refrigeration operation values of 1,156,000 kwhrs per year assigned to the single-duct primary system and 829,000 kwhrs per year assigned to the dual-duct system and using 2 cents per kwhr the apparent savings is \$6,540 per year. This still leaves the overall operating cost for the single-duct primary system considerably lower.

It is stated that the dual-duct one-fan system is generally lower in initial cost than the single-duct primary system. We know of no basis for making this estimate since direct comparisons are infrequent but in several instances that the writer is acquainted with, when comparisons were attempted, they did not substantiate the author's statement. Moreover, no first cost comparison is valid unless it includes evaluation of relative space requirements.

The statement in the paper that the risers in a single-duct primary system will occupy 2.5 percent of the building floor area, whereas the double-duct risers would occupy 3 percent has not been checked, but even this difference of  $\frac{1}{2}$  of 1 percent on a building of 238,000 sq ft represents \$24,000 when one considers that it costs about \$20 per sq ft to build an office building. In addition, the fan rooms on a dual-duct system would either be considerably larger or more of them would be required because as indicated, this system must handle three or more times as much air, and return air fans as well as primary air fans are effected.

In addition, there is a statement that operating and maintenance supervision is 25 percent higher for the induction unit system than the dual-duct system. This too would be difficult to substantiate, but over a long period of time, through a very large number of jobs, costs for cleaning, and for maintaining water valves on single-duct primary systems are available. These are accurate costs kept by building operators. It would be interesting to compare these with comparable figures which the author might have for large double-duct systems operating over many years.

M. G. KERSHAW, Philadelphia, Pa., (WRITTEN): The author is complimented on his chart presentation of the direct effect on heating and refrigeration capacity required for various ventilating loads per square foot of occupied building area. With charts of this type management may more readily select a system with sufficient flexibility and capacity to provide for future changes in anticipated people loading.

The writer is familiar with two-pipe ventilating and heating systems as installed in two- or three-story schools but has not observed the operating results of this type of system as applied to comfort conditioning.

We would like the author's comments on the following: (1) It appears that the *dual-duct* supply system would occupy more rentable floor space than the *single-duct primary* high velocity system. Does the author have a slide showing comparison of required space? (2) Annual operating costs per square foot as given in Table 1 appear higher than those which have been mentioned in recent articles on modernization of office buildings. These figures are always subject to misunderstanding because of varying factors such as load costs and different management policies. The writer raises this point as he understands the author indicates the system he describes would have lower operating cost. My comparison did not confirm this point. (3) How does the

author combat the objection to cross zone contamination with recirculating of the supply air? (4) Would the author describe details in the design of outlet which insure against stratification of the two air streams?

S. R. LEWIS, Chicago, Ill., (WRITTEN): This problem and its various solutions will be encountered more and more; especially as refrigerated cooling is applied to existing buildings. I commend the author's equitable analysis.

The single-duct primary air system proves admirable for exterior room, but due to rapid changes in outdoor temperature in cities like Chicago, it requires an appreciable length of time and some skill by the operator to change the water temperature in the room-unit coils from re-cool to re-heat. Interior spaces usually require separate treatment.

The dual-duct system is the logical sequence of the hot deck-cold deck heating and ventilating plants long used in school buildings, where each classroom has a thermostat which adjusts a double blade damper to select air supply at a temperature either warmer or cooler than that at the thermostat. The dual-duct system proves more costly for duct, but reduces the investment in water piping and is instantly available in winter or in summer.

Based on cooling of old hotel guest rooms it appears that if one must install air ducts from central air supply systems, two separate air supply ducts above the corridors can be furred in about as easily as can a single air supply duct.

I have not found serious drafts or noise due to variation in air volume delivered to each room, following changes in the double damper adjustments in the other rooms in a particular zone.

I know of no acceptable experience with a high pressure central air supply system in which individual temperature control is obtained by intermittently readjusting the amount of air delivered into each room.

Room unit systems in which each room has a fan, coil and filter, recirculating the room air and mixing with this air a small percentage of outside air admitted through a hole in the wall, probably involve the lowest investment for an old building. The coils in the units receive hot water in winter and chilled water in summer. In intermediate weather which can happen at any time, the operating engineer may be unable to foresee whether chilling or reheating will be required and which ever may be needed, can not be delivered instantly. Experience amply justifies the author's objections to the room-unit scheme.

Table 2 is valuable as applied to an office building.

My friend, E. P. Heckel, currently is participating in proposed revisions of the Chicago Ventilation Code, and he makes the following comments:

(1) The ventilation codes of some cities, such as Chicago, require that more than 0.2 cfm of new air per square foot of floor area, shall be supplied by a mechanical air conditioning ventilating system.

(2) The suggestion of 80 F temperature with 50 percent relative humidity as the optimum indoor conditions for human comfort, during the summer months, may call for some reconsideration and perhaps, revision of the ASHAE comfort charts. More recent observations tend to indicate that 76 F dry bulb temperature and relative humidity reaching up to 65 or even 70 percent is conducive to the comfort of many people, even though the outdoor temperature may be higher than 95 F.

F. W. CHAMBERS, Toronto, Ont., Canada, (WRITTEN): I believe that the author of this paper has done a very worthy piece of work in outlining the several methods of air conditioning which are used in multi-room buildings. In this connection, I call attention to the point that the dual-duct one-fan system which Mr. Waterfill mentions is covered by a Canadian patent No. 457,313. I believe that this point is of interest to engineers who are proposing to design such a system for installation in Canada because of the necessity of obtaining a license under the patent.

R. P. COOK, Rochester, N. Y., (WRITTEN): This extremely practical and very thought-

ful paper should be welcomed by many designers of air conditioning systems, particularly by those of limited experience. It illustrates the desirability of careful consideration of various systems prior to final design. I have a feeling that many designers do not actually design, calculate and compare the economics of several systems before deciding on the type of system to be used.

One important factor which the author has not included but which is frequently overlooked by the designer, is the technical qualifications of the average operating superintendent. A system may be carefully designed to be economically capable of meeting every change in weather and every change in internal load but lack of skillful operation can result in high operating costs. Almost every change in operating conditions can, theoretically, be adjusted for through elaborate automatic control devices but even these can fail to justify the investment if they are not thoroughly understood and carefully maintained by the operating personnel. An operator with proper training in the essential theory of air conditioning can operate a properly designed system at maximum efficiency but, unfortunately, such operators are not always available. The designer must therefore consider the probabilities of good operation in his initial design. Sometimes simplicity of operation is more important than theoretical perfection. The designer should remember that he, with all of his knowledge of the whys and wherefores of his design, will not be the operator. Certain rules for operation can be formulated for the operator to follow. Beyond these rules, economical operation will depend on the intelligence and skill of the operator and the complexity of the original design. A large installation, such as that used as an example by the author, probably will be operated more intelligently than a small installation where the owner may not feel justified in paying the salary of a high grade operator. The designer should try to determine as best he can the type of operation which his system may expect and this should, to some extent, influence the trend of his design; a choice between maximum efficiency under good operation or reasonable efficiency from a system which can be operated largely by a set of rules.

**AUTHOR'S CLOSURE:** I will consider first those written discussions received prior to the meeting.

The comments by Mr. Cook on the design and the economic comparisons and analysis of operating skill and labor requirements are pertinent and well stated. The simplification of operation is always desirable and a decision should be made on the basis of complete analysis of all factors.

I appreciate Mr. Lewis' approval of the contents of the paper, because I feel that his position in the industry is outstanding.

The instant availability of warm or cold air to any space at any season, as provided by the dual-duct method, as he states, seems basic for best results in multi-room buildings.

Mr. Heckel very properly calls attention to the ventilation codes in cities and to the current comfort thinking. I have seen no scientific or experience data to justify some of the trends in high ventilation rates. From the standpoint of CO<sub>2</sub> and body odors, existing data indicate that sixteen cfm per person provides a high margin of safety and air purity. For heavy smoking areas some increase may be justified, although a recent paper by Professor Yaglou indicates that when put on the basis of the smokers present, sixteen cfm per occupant would be satisfactory in a majority of cases.

The chief objection to excessive ventilation is that while it accomplishes very little that we can prove, it is very expensive in installation and operating costs.

In the matter of current comfort thinking, for many years 80 F dry bulb and 50 percent relative humidity was considered a good design condition. According to the ASHAE comfort chart, 76F and 65 percent relative humidity represent the same effective temperature for persons fully clothed, at rest and in still air.

My own personal observation is that 76 F is too cool for summer clothing and light activity while 65 percent is too humid for real work. 80 F dry bulb and 40 to 50 percent humidity is generally more invigorating. High humidities are stifling.

Mr. King requests more data on construction and financing. That is quite a large order which I hope someone will fill at another date.

I would like to expand on the dual-duct system further, but time at the moment is limited.

Mr. King asks clarification of the statement about heat transfer from an occupied room to an unoccupied room. If room temperatures are controlled at 80 F during the cooling season, then the return air temperature will be 80 F by virtue of the heat from the occupied rooms. This return air is available in any quantity required by an unoccupied or lightly loaded room, thus maintaining complete temperature control. The humidity is maintained in relation to the total load as shown in Figs. 8 and 9. Ventilation too is maintained by the minimum outside air setting.

Mr. Kershaw speaks of the rentable space occupied by the two systems. In either case it is very small and from all I can determine the difference between them is almost negligible. The dual-duct probably occupies slightly more space, less than half of one percent of the rentable area under average conditions, but the advantage could reverse especially where large conventional systems comprise an appreciable component of the total installation.

The operating costs seem higher to him than in previously published figures. That is entirely possible because this analysis is based on a ten-hour day and on maximum occupancy during that time. This is one means of setting up a comparable basis. It is easily modified for specific conditions. If a nine-hour day is used instead of a ten-hour day, operating hours and costs are reduced.

Mr. Kershaw also asks how one combats cross contamination with recirculated air. In a large auditorium such as we now occupy, the air is in constant motion. There is cross contamination from side to side equivalent to recirculation from many cubicles, yet ventilation in terms of occupancy is the only question considered here. We will have to live in isolated cells if we are going to avoid completely cross contamination. Ventilation in terms of occupancy seems the only rational consideration, and purity is high under present practice.

Stratification is avoided in two ways, first by mixing the warm and cold streams in the unit before discharge, and second a relatively high outlet velocity induces room circulation and mixing.

I agree with Mr. Giannini that this is a broad and exceedingly important topic, which justifies and will in time receive broader treatment. This paper is limited to basic analyses and is presented with the intent of rectifying many false opinions that have developed.

The single-duct primary system must be zoned so that all rooms on a given air and water supply circuit have substantially the same load cycle. With this basic design, individual room control is reasonably reliable and many installations show this is practical. However, there are occasional pattern deviations beyond the flexibility of the system. Scheduling reheat of the primary air as a function of outside air is a sound means of staying within the system limits of flexibility. Throttling the water control valve is effective within its range, but at certain seasons the water temperature must be changed from heating to cooling and back to heating all in the same day.

Figs. 6, 7, 8 and 9 are strictly psychrometric analyses as indicated in the paper. If there is no overcooling or excessive humidity in an installation, it simply means that sensible heat loads or reheat exist as shown. These analyses are for peak outdoor conditions. The lighter loads usually occur when outside dewpoints are lower, consequently both primary and dual-duct perform better at light loads in actual operation.

As the load decreases in any room with the dual-duct one-fan system, a greater percentage of warm air is admitted, with the result that the humidity of that room shifts from curve A and approaches C, a small increase. Any contention that the humidity in this room must build up beyond these valves is unsound. The formulae in the appendix determine the humidity conditions quite accurately, in an engineering manner. These formulae do not imply a steady state condition any more than do thermal load

calculations. They are applicable in determining conditions thruout the conditioned spaces for any combination of sensible and moisture loads.

Mr. Giannini is quite correct in stating that static unbalance is a real problem in dual-duct installations. It is this problem which has delayed acceptance of the dual-duct system. Some engineers having recognized the many other advantages of the dual-duct system are developing effective static pressure or volume control means, which do not impair the good qualities of the system.

The paper does not "point out that a two-fan system is required to minimize the amount of undehumidified outside air entering the system", as such is not the case. A dehumidifier in the outside air can easily accomplish any degree of dehumidification desired, though seldom required and not used in the analysis of Figs. 8 and 9. The two-fan system is an economical means of dehumidifying large volumes of outside air requiring less reheat than either the single-duct primary system or dual-duct one-fan system. Ventilation is maintained in the two-fan dual-duct system by reheat sufficient to overcome the excess cooling effect of outside air when internal sensible heat loads are low. The warm air temperatures required for this purpose are around 90 F. There is no conflict here with Mr. Wilson's paper.

The performance of the typical office building is an analysis based on a typical set of engineering conditions. As stated in the paper, the analysis holds whether periphery or interior, as long as loads are as stated. It is recognized, however, that the single-duct primary is not suited to interior zones. The comparison may be between systems applied to the periphery only, even though the dual-duct is suited to both interior and exterior. If one chooses to get technical on this point, the comparison may be limited to periphery only. The interior could be handled by the same method in both cases, keeping the comparison clear of the confusion created by introducing other systems either in the same building or elsewhere.

Mr. Giannini makes some changes in temperature ranges and primary air volumes, Table 1 Building 2, to build up the ratio of air volumes handled by the fans of the dual-duct to single-duct primary systems, but does not follow through on other resultant effects. With 0.4 cfm per sq ft of primary air, the sensible cooling capacity of the primary is reduced to 10 Btu per sq ft and the secondary circulation must be increased to almost 4:1. This requires increased static pressure, increased noise in the room and an increase in HP per cfm of primary air. Furthermore, the system requires more refrigeration at intermediate seasons because the use of favorable outside air is further restricted. Other effects would also require consideration. Why an additional 2 deg F is considered is not clear, as the duct rise has already been included in the paper in estimating refrigeration operating costs.

The paper was prepared on an assumed internal temperature of 80 F. The analysis could have been made on any other temperature or could have included a schedule of reduced indoor temperatures, based on outdoor changes or on preferences of individual tenants. This would not, however, necessarily upset the relative behavior of the systems materially. Both systems would require increased air supply or lower fluid temperatures for lower room temperature at maximum conditions or for dropping the indoor temperature to 76 F when the outside reaches 85 F.

It may be possible to cool Building 2 with 0.4 cfm per sq ft primary air, but it would be dubious engineering, because the primary air circuit would not be doing a reasonable proportion of the cooling and the use of favorable outdoor air would be seriously restricted. This increases the need for chilled water for a longer period, probably thruout the year.

The refrigeration demand chart (Fig. 10) was plotted for Building 1 where the primary air sensible heat capacity is  $11/25 = 44$  percent of the room load. A primary supply of 0.4 cfm for Building 2 has a sensible heat capacity of less than 30 percent for an 80 F dry bulb room and less than 25 percent for a 76 F dry bulb room. Fig. 10 is, therefore, low for a 0.4 cfm primary because favorable outside air is more restricted in relieving the mechanical refrigeration load. A comparable primary supply would be  $(44\% \times 35) \div (0.074 \times 0.24 \times 25 \text{ deg} \times 60) = 0.58 \text{ cfm per sq ft}$ .

Referring to Fig. 5, it is noted that 0.4 cfm per sq ft outside air at 50 F wet bulb decreases the mechanical refrigeration 0.0014 tons per sq ft, whereas 0.58 cfm per sq ft decreases it 0.00205 tons per sq ft. The difference 0.00065 tons per sq ft must then be added to the mechanical refrigeration load at this mean condition for the 0.4 cfm primary. This is an average increase of 155 tons and at the averaged rate of 1.10 kw per ton for the increased hours at the lower capacities, represents an average power increase of 170 kw. The refrigeration load would increase beginning at about 65 F wet bulb and extend beyond the 35 F limit of operation of Fig. 10, increasing the operating time fully 5 percent. Since this range represents 85 percent of the revised operating hours, the refrigeration kwhrs would be increased  $170 \times 0.85 \times 1726 \times 1.05 = 261000$  kwhrs. The total for these items for 80 deg internal temperature for both systems, therefore becomes:

For single-duct primary system (assuming no increase in static pressure for increased ratio of induced circulation to primary or increased refrigeration kw demand and supervision charges):

Fans and secondary pumps	$260 \text{ HP} \times 0.9 \times 0.02 \times 2600$	$= \$12,200$ per yr.
Refrigeration and Cool. T.	$1,417,000 \text{ kw hrs.} \times 0.02$	$= 28,340$ per yr.
	Total	<u>\$40,540</u> per yr.

For the dual-duct system, corrected to 1.12 cfm supply,

Fans	$459 \text{ HP} \times 0.9 \times 0.02 \times 2600$	$= \$21,500$ per yr.
Refrigeration and Cool. T.	$829,000 \text{ kw hrs} \times 0.02$	$= 16,580$ per yr.
	Total	<u>\$38,080</u> per yr.

Fig. 11 applies acceptably to Building 2 because full advantage is taken of favorable outside air as in Building 1.

Equipment floor area requirements are variable even for the same type system and, where combination systems are employed, the percentages could change in favor of the dual-duct. However, cramming equipment of this value into the minimum possible space, where maintenance is difficult, is one of the outstanding phobias of building owners. Such equipment without electrical and structural changes, will cost the owners over one million dollars, yet basement, roof and corner spaces are restricted to trim the allotted space for a saving of a few thousand dollars. The  $2\frac{1}{2}$  percent and 3 percent of floor area, as stated in the paper, includes all apparatus space and not just the air and water risers as Mr. Giannini interprets it.

It is difficult to arrive at a clear cut comparison of operating and maintenance supervision costs. However, a study of the servicing of dispersed coil units and water valves versus central apparatus, plus discussions with building operators, resulted in the conclusion that the 25 percent higher cost for the single-duct primary is reasonable.

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## EFFECTS OF WEATHER CONDITIONS ON COOLING UNIT OPERATION IN A RESIDENCE†

By H. T. GILKEY\*, W. F. STOECKER\*\*, AND S. KONZO‡, URBANA, ILL.

AS A PART of the cooperative project sponsored jointly by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*, investigations of summer cooling were conducted in Warm Air Heating Research Residence No. 2 during the summers of 1952 and 1953. A portion of the investigation conducted in Research Residence No. 2 during the summer of 1952 has been reported previously<sup>1</sup>.

The principal objectives of the investigation reported in this paper were:

1. To determine the effect of continuous and cyclic blower operation upon the air temperature and humidity conditions in the Residence.
2. To determine a rational basis for estimating the hours of compressor operation during the summer season.
3. To determine the velocity and temperature patterns of the air stream issuing from a high wall supply register.
4. To determine the effect on compressor performance of increasing the air-flow rate from 300 cfm per rated ton to 400 cfm per rated ton of refrigeration.

### RESEARCH RESIDENCE No. 2

The Residence, shown in Fig. 1, is a one-story structure of frame construction with a large amount of glass exposure and with a full basement. The calculated coefficient of heat transmission,  $U$ , for the insulated exposed wall section is 0.07 Btu per (hr) (sq ft) (F deg). All windows and doors are weather-stripped and the windows on the east and west exposures are equipped with canvas awnings. Except for one large picture window in the living room, which was fixed in place and consisted of two panes, the windows are single glazed and of the horizontal sliding type. Doors are of conventional wood and glass construction. The south exposure of the Residence is shaded by a 3-ft 10-in. roof overhang. For this

† This investigation was a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*.

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<sup>1</sup> Exponent numerals refer to References.

Presented at the 61st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Philadelphia, January 1955.

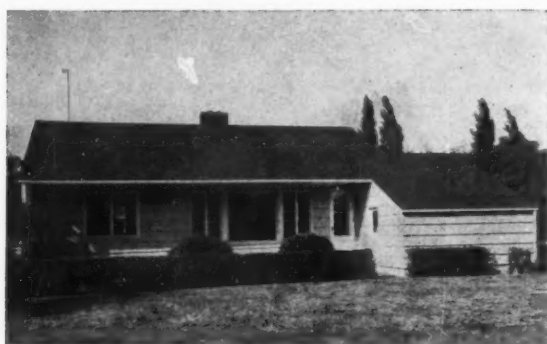


FIG. 1. SOUTH VIEW OF RESEARCH RESIDENCE NO. 2

summer cooling investigation, the ceiling joist space was filled with 5 in. of mineral wool insulation, and the calculated coefficient of heat transmission is 0.07 Btu per (hr) (sq ft) (F deg).

TABLE 1—DATA FOR RESEARCH RESIDENCE NO. 2

HEAT TRANSMISSION COEFFICIENTS, BTU PER HR (SQ FT) (F)					U			
INSULATED FRAME WALL, WITH 3½-IN. MINERAL WOOL INSULATION					0.07			
INSULATED CEILING, WITH 5-IN. MINERAL WOOL INSULATION					0.07			
OUTSIDE DOOR—FRONT HALL					0.51			
ROOM	DIMENSIONS		CEILING AREA <sup>b</sup>	EXPOSURE	NET WALL AREA	GLASS AREA	VOLUME	
	FT IN.	FT IN.						
Living Room	21	10 × 13	4	308	East	117	—	2,480
South Bedroom & South Bedroom	13	4 × 11	0	169	South	89	100 <sup>b</sup>	1,325
	4	0 × 2	3		West	71	28	
Closets (2)	4	6 × 1	11	23	South	19	—	147
Bath	7	8 × 4	11	44	West	36	11	320
North Bedroom	11	11 × 10	4	135	West	66	25	1,050
					North	61	30	
North Bedroom Closet	5	10 × 2	4	18	North	24	—	116
					East	37	—	
Hall to Bath	6	7 × 5	2	43	—	—	—	289
Front Hall	11	6 × 4	7	59	North	22	21 (door)	448
Front Hall Closet	4	0 × 2	4	9	—	—	—	79
Kitchen-Dinette	19	0 × 11	4	232	North	124	42	1,830
					East	89	12	
Total First Story				1,040	—	848	273	8,084

<sup>a</sup> Ceiling area includes area of partition and exterior walls.

<sup>b</sup> South Living Room glass area includes area of outside door having a high glass to wood area ratio. Double glazed picture window has area of 27 sq ft.

Note: Ceiling Height of First Story—8 ft 6 in.

The conditioned space consisted of all first-story rooms. Table 1 gives a summary of the room dimensions and volumes. The heat gain calculations, made in accordance with Manual No. 11<sup>2</sup> of the *National Warm Air Heating and Air Conditioning Association*, were based on outdoor design conditions of 95 F dry bulb and 76 F wet bulb, and an indoor dry-bulb temperature of 75 F. The design sensible heat gain on a design day for the structure was 21,090 Btu per hr. The residence was completely furnished but unoccupied during this investigation.

**Cooling Unit:** The cooling unit, shown in Fig. 2, was one of two sections of a year 'round air conditioner which was installed in the basement and described in a previous paper<sup>1</sup>. Return air entered the unit and passed downward through the filter and evaporator coil, both of which were in a horizontal plane. The

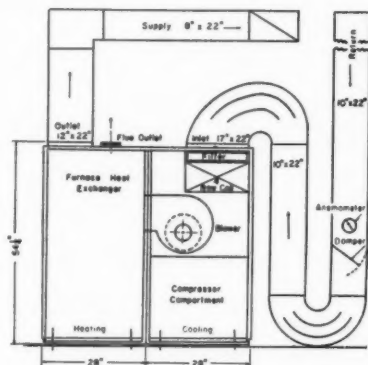


FIG. 2. FURNACE AND COOLING UNIT

pan for collecting the water condensed out of the air was located beneath the evaporator coil. The coil and blower were in an insulated compartment and were isolated from the compressor and condenser, which were located in the lower part of the unit. Refrigeration capacity of the unit was 23,800 Btu per hr with ASRE standard inlet-air conditions and at a rated air delivery of 900 cfm and 120 psi refrigerant head pressure. The semi-hermetic compressor was directly connected to the 220 volt, single-phase, 2 hp motor. Both the motor and the shell-and-tube type condenser were water cooled. Refrigerant was dichlorodifluoromethane.

**Duct System:** The duct system was the same used previously in both heating and cooling investigations; it was of the extended plenum type, having uniformly sized trunk ducts leading from the furnace bonnet toward the east and west ends of the basement. Branch ducts were connected to the top or side of the trunks and were unchanged in size from the trunk take-off fitting to the register stack-head. All registers in the first-story rooms were at the high-sidewall location, 6½ ft from the floor, with the exception of the baseboard register in the front hall near the door. Fig. 3 shows the first-story plan and the register locations. All return-air intakes were located in the baseboard. The system was designed

for heating only in accordance with Manual No. 7<sup>3</sup> of the *National Warm Air Heating and Air Conditioning Association*.

**Instrumentation:** Approximately 180 thermocouples of 24-gage copper-constantan wire were installed for temperature measurements. Thermocouples were placed at four different levels (floor, sitting, breathing, and ceiling) on standards located near the centers of each of the first-story rooms and at three stations in

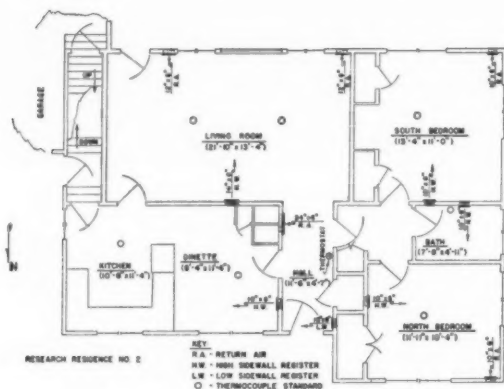


FIG. 3. FLOOR PLAN OF RESEARCH RESIDENCE NO. 2

the basement. The locations of the temperature measuring stations in the first-story rooms are shown in Fig. 3. Thermocouples were also installed in the ceiling and floor surfaces, on the surfaces and within the exterior walls, in the attic, in the duct system, and at other desired points inside and outside the Residence. Each thermocouple was connected through switches to an indicating potentiometer. It was possible to obtain a continuous record of any 18 of the 180 thermocouple stations by means of recording potentiometers.

Heat-flow meters were installed at various locations in the house. One meter was installed in each of the outside walls 60 in. from the floor, and was located on the exterior surface of the interior panels and next to the insulation. One was placed above the ceiling, between the top ceiling surface and the attic insulation. Meters were also located on the underside of both the north and south slopes of the roof.

Continuous records were made of the outdoor-air temperatures, both dry and wet bulb, the indoor relative humidity, the wind velocity, and the intensity of solar radiation received on a horizontal plane. Wind direction was noted at frequent intervals.

#### OPERATING CONDITIONS

Three series of studies were conducted, all with an air-flow rate of 800 cfm, or approximately 400 cfm per rated ton of refrigeration. This was in contrast with 300 cfm per rated ton used in the studies<sup>1</sup> of 1952. No ventilation air was me-

chanically introduced during any of the 1953 studies, and the Residence was kept closed at all times. The indoor dry-bulb temperature was controlled at 75 F, but no attempt was made to control the indoor relative humidity.

The three series of studies conducted in 1953 were identified as follows:

*Series S53-1:* The blower was operated continuously. The Residence was not occupied and no moisture was released indoors.

*Series S53-2:* The blower was operated continuously. For the purposes of imposing a definite latent load, the following weights of water were vaporized by means of electric vaporizers: 1.0 lb in 30 min. beginning at 7:00 a.m., 1.25 lb in 1 hr beginning at 11:30 a.m., and 2.25 lb. in 2 hr beginning at 5:00 p.m.

*Series S53-3:* The blower was cycled with the compressor. The latent load addition was the same as for *Series S53-2*.

### EFFECTS OF BLOWER OPERATION

*Increase in Air-Flow Rate:* In 1952 the measured capacity of the condensing unit was 20,300 Btu per hr with an air-flow rate of 300 cfm per rated ton of refrigeration. In these studies, a 33 percent increase in the flow rate of the circulating air to 400 cfm per ton resulted in a measured capacity of 21,700 Btu per hr, a 7 percent increase. The fact that the measured capacity was still below the rating of the unit was attributed largely to the return-air temperature, which was lower than that used for rating the condensing unit. Other deviations in operating conditions from ASRE standard conditions caused the remainder of the discrepancy.

With an air-flow rate of 800 cfm the total pressure in the supply plenum at the exit of the year 'round conditioner was 0.21 in. of water column and that at the return-air plenum was 0.19 in. of water column, or 0.40 in. for both sides. These values are approximately double those used in the original design of the duct system. An air-flow rate of 340 cfm was used to heat the first story only. When the basement was heated, an air-flow rate of 565 cfm was used for both the basement and the first story, and the extended plenum was sized for this condition.

*Indoor Humidity Ratios:* Although higher outdoor humidity ratios can be expected to occur with higher values of mean daily outdoor temperature, the variation which exists is indicated by the spread of points on the left hand plot of Fig. 4. In general, the trend of the observed data shows that on hot days a higher moisture load can be expected than on cool days. The points indicated by the solid circles show that for a majority of the 15 days during which the blower was cycled with the compressor (*Series S53-3*), the outdoor air was relatively dry.

The plot on the upper right portion of Fig. 4 shows that on hot days more moisture was removed by the evaporator coil than was removed on cool days. The solid circle points, representing data for cyclic blower operation, were predominantly near the upper range of observed points. Thus, in spite of the fact that the tests were conducted during a dry spell, the total amount of moisture removed was relatively large. The majority of the points represented by the open circles (*Series S53-2*) were higher than those represented by crosses (*Series S53-1*). This would indicate that the release of 4.5 lb of moisture within the house (*Series S53-2*) resulted in a larger amount of condensation during the day.)

Indoor humidity, as represented by the humidity ratio, is shown by the plot on the lower right of Fig. 4. For both series of tests during which the blower oper-

ated continuously (*Series S53-1* and *S53-2*), the indoor humidity ratio was practically constant regardless of the outdoor temperature. This average humidity ratio was about 0.0123 lb of moisture per lb of dry air and corresponds to a relative humidity of 66 percent at a temperature of 75 F. In other words, with larger moisture loads arising either from more humid outdoor air or a larger moisture release indoors, a larger amount of condensation occurred.

*Blower Cycled with Compressor:* Although the most common method of blower operation has been to operate the blower continuously, the suggestion has been made that better humidity control might be achieved if the blower were cycled with the compressor. The underlying theory is that if the blower ceases to operate when the compressor stops, the moisture condensed on the cooling coil will not

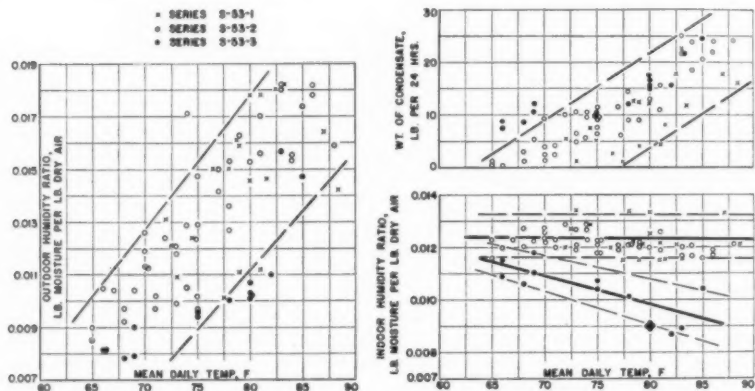


FIG. 4. HUMIDITY RATIOS AND WEIGHT OF CONDENSATE

be re-evaporated. The humidity of the room air would not be increased by this re-evaporation, as it would when the blower is operated continuously.

The data shown in the lower right side of Fig. 4 for cyclic blower operation (*Series S53-3*) indicated that on a hot day the indoor humidity ratio was lower than on a cooler day. This may only be accounted for by the longer operating time of the compressor on a hot day, with the consequent larger condensation of moisture, and the absence of re-evaporation from the wet coils during periods when the compressor and blower were not operating. The difference between the curve representing continuous blower operation and that for cyclic blower operation is sufficiently large that it cannot be attributed merely to differences in weather conditions during the two studies. It might be expected that the indoor humidity ratio was lower for this type of operation as a result of the lower outdoor humidity ratio experienced during this series. It should be noted, however, that the weight of condensate collected during the period with cyclic blower operation was higher than the mean for the entire investigation, but that on days of high mean daily temperature the weight of condensate was within the range experienced with continuous blower operation. The practical implication of the results is that cyclic operation of the blower in conjunction with the compressor gives a larger reduc-

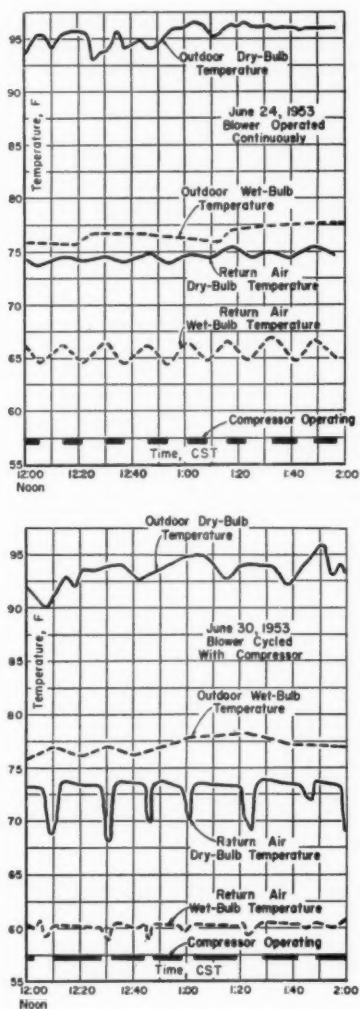


FIG. 5. INDOOR AIR CONDITIONS WITH TWO METHODS OF BLOWER OPERATION

tion in humidity ratio than does continuous operation, especially in hot weather. Nevertheless, it would be expected that the weight of condensate collected would be the same for the two methods of blower operation for identical days on which the compressor operated continuously.

A comparison of the two methods of blower operation is shown in Fig. 5. The examples shown are from noon to 2:00 p.m. on two days when similar outdoor temperatures and humidities were experienced, as shown by the outdoor dry-bulb and wet-bulb temperatures. The indoor air conditions were measured in the return-air plenum at a point just upstream from the point where the air entered the cooling unit. With continuous blower operation, shown on the left side of Fig. 5, the indoor wet-bulb temperature increased slightly with each compressor cycle. With cyclic operation of the blower, shown on the right side of Fig. 5, considerably lower return-air wet-bulb temperatures were experienced. The sharp decreases in dry-bulb and wet-bulb temperatures, shown on the right hand charts, after the blower stopped operation have been attributed to the proximity of the recording thermocouples to the cooling coil, although an air filter did serve to shield the thermocouples from the cool evaporator coil.

Although cycling the blower with the compressor would seem to be desirable from the standpoint of lower indoor humidity and lower operating cost of the blower, the possibility of discomfort due to lack of air circulation during the off-periods of the blower must be considered. Unfortunately, this is a subjective matter, which is difficult to evaluate in this type of investigation.

#### VARIABILITY OF WEATHER

From a practical standpoint the most difficult estimate that must be made for the summer cooling of residences is the number of hours of operation of the cooling unit. Once such an estimate is established for a given residence, the electrical power consumption and water consumption can be readily computed. In view of the interest in the subject, a brief discussion is presented of the inherent obstacles confronting those attempting to arrive at a reasonable estimate of the operating times for a season.

In previous publications<sup>5,6</sup> extreme variability of the summer weather was shown to exist. For example, during the four-year period of 1935 to 1938, the values given in Table 2 were tabulated from the weather records at Research

TABLE 2—WEATHER RECORDS FOR THE YEARS 1935 THROUGH 1938<sup>a</sup>

ITEM	SEASON			
	1935	1936	1937	1938
Number of Days at 85 F or over.....	55	81	61	54
Number of Days at 90 F or over.....	19	50	23	16
Total Hours above 85 F.....	23	706	302	204
Total Hours above 90 F.....	83	342	40	23
Total Degree-Hours above 85 F.....	1082	4551	862	535
Total Degree-Hours above 90 F.....	174	1963	64	23

<sup>a</sup> From weather records obtained at Research Residence No. 1 in Urbana, Ill.

References: University of Illinois Engineering Experiment Station Bulletins No. 305, p. 64, and No. 321, p. 22.

Residence No. 1. It is true that the summer of 1936 was the hottest summer in the history of the local weather bureau, but in more recent years the summers of 1952, 1953, and 1954 have approached the severity experienced in 1936. Regardless of the indices used to express the severity of the summer weather, the fact remains that the numerical values of the indices did vary more than 100

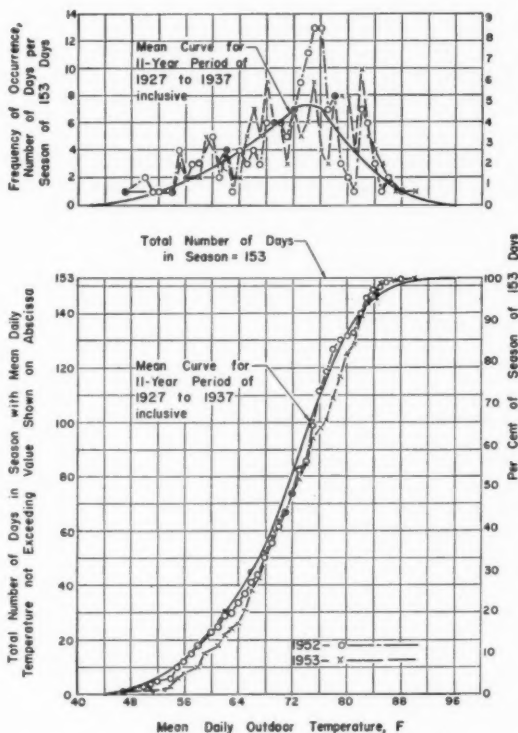


FIG. 6. FREQUENCY OF OCCURRENCE OF DAYS HAVING STATED MEAN DAILY TEMPERATURES

percent from the maximum values. This is in decided contrast with winter heating experience, in which the number of degree days for any season does not vary more than about 25 percent from the maximum values, and not more than about 15 percent from the mean value for the locality. In other words, estimates of fuel consumption for the heating season based on the mean value of the degree days can be made with some assurance that the estimates will be reasonable for at least three years out of four. For summer cooling, however, where the indices of weather severity can vary as much as 100 percent or more from one season to the next,

some question exists whether the mean value for a period of years will be entirely satisfactory. The suggestion is offered that a statistical analysis should be made and that any value finally selected for a given locality should not be exceeded more often than one summer out of four. No such analysis has been made for any locality, primarily because no agreement has been reached on the index to be used for evaluating the summer weather.

A comparison of the weather experienced during the 1952 and 1953 cooling seasons with the mean conditions experienced from 1927 to 1937 inclusive is shown in Fig. 6. For example, in 1953 there were a total of 10 days on which the mean daily temperature was 82 F. In a season typical of the 11-year mean, however, one would find only three days having this mean temperature. It can be noted

TABLE 3—WEATHER DATA AND ESTIMATED COMPRESSOR OPERATION  
(Based on 153-day period from May 1 to September 30, 1953).

	MAY	JUNE	JULY	AUG.	SEPT.	TOTAL
<i>Weather Data</i>						
Number of days at 85 F and above <sup>a</sup>	6	21	22	18	12	76
Number of days at 90 F and above <sup>a</sup>	3	17	11	11	7	49
Number of days at 95 F and above <sup>a</sup>	0	5	3	9	5	22
Number of days of mean daily temperature 65 F and above	13	30	31	31	22	127
Number of degree days for cooling <sup>b</sup>	94	372	352	305	149	1,272
<i>Estimated Consumption with Cyclic Blower Operation</i>						
Estimated compressor Operation, hr	58	205	197	175	92	727
Estimated electrical consumption, kwhr						
Compressor	116	410	394	350	184	1,454
Blower <sup>c</sup>	19	66	63	56	29	233
Total	135	476	457	406	213	1,687
Estimated water consumption, gal	8,600	30,300	29,100	25,900	13,600	107,500
<i>Estimated Consumption with Continuous Blower Operation</i>						
Estimated electrical consumption, kwhr						
Compressor	116	410	394	350	184	1,454
Blower <sup>d</sup>	100	230	238	238	169	975
Total	216	640	632	588	353	2,429
Estimated water consumption, gal	8,600	30,300	29,100	25,900	13,600	107,500

<sup>a</sup> The temperatures refer to maximum daily temperatures, and not to mean daily temperatures.

<sup>b</sup> Degree days per day equals the difference between the mean daily temperature and 65 F. Mean daily temperatures below 65 F are not considered.

<sup>c</sup> The blower electrical consumption is based on cyclic blower operation during compressor operation only.

<sup>d</sup> The blower electrical consumption is based on 24-hr blower operation on days in which mean daily temperature exceeds 65 F.

further from the lower curve in Fig. 6 that the number of days during the 1953 cooling season upon which the mean daily temperature did not exceed 65 F was 31, and that this mean daily temperature was exceeded on 122 days. Thus, as will be shown later, the compressor would have operated on 127 days (122 days during which  $t_m > 65$  F, plus 5 days during which  $t_m = 65$  F) out of the 153 days from May 1 to September 30, 1953.

### INDICES OF WEATHER

In earlier studies made in Research Residence No. 1 during the years of 1932 to 1939, numerous attempts were made to correlate the number of hours of compressor operation with some index of weather condition. Obviously, as the daily outdoor temperature increases, longer hours of compressor operation can be expected. The problem is not as simple as stated, even for a given residence, since no single index of weather conditions can take into account such variables as: (1) intensity and duration of maximum outdoor dry-bulb temperature; (2) intensity of outdoor dry-bulb temperature during the hours preceding the maximum outdoor temperature; (3) magnitude of the outdoor humidity ratio during the day; (4) the duration and intensity of solar radiation; and (5) moisture release and heat release within the residence. In spite of the difficulties, however, it becomes of great practical interest to determine some relatively simple index of weather conditions that might serve as a correlation for compressor operation.

The number of degree hours above 85 F, shown in Table 3, was used in the earlier studies, and was found to be one index which could be used to correlate the number of hours of plant operation. The differences in the constructions of Residence Nos. 1 and 2, as well as the lower setting of the room thermostat now being recommended, have shown that a lower base temperature than 85 F should be used for determining the degree hours. For example, in the early thirties, when thermal shock was considered to be a possible source of discomfort, the setting of the room thermostat was at the upper limit of comfort, namely between 78 F and 80 F. In current practice, a setting of 75 F is commonly recommended. This small difference of 3 F in the setting of the room thermostat will affect to a considerable extent the number of hours of compressor operation, since in general, the rate of rise of room-air temperature in a residence may be as low as  $\frac{1}{2}$  to 1 deg per hr without cooling. Furthermore, the rooms in a one-story house of current construction with relatively larger glass area, will require more cooling and at an earlier hour than the first-story rooms of a multiple-story residence with smaller glass area. In addition, the extent to which night-air cooling is employed will also affect the starting time of the cooling, and hence the total hours of operation each day.

### COMPRESSOR OPERATION

The studies conducted during the summer of 1953 are summarized in the three plots shown in Fig. 7. The lower set of curves shows the relationship between the hours of compressor operation and the number of degree hours above a base temperature of 65 F for each day. The degree hours were determined only for the period during which the outdoor air temperature exceeded the 65 F value for each day. In other words, unlike the degree day determinations for winter heating, the mean daily temperature for the day is not used in determining the degree hours above 65 F. The lower set of curves in Fig. 7 shows good linear correlation

between compressor operation and degree hours above 65 F, perhaps better than the correlations obtained with either the maximum daily temperature or the mean daily temperature, shown in the two upper sets of curves. From the standpoint of industry-wide application, however, the degree hour index is not as convenient as one based on a simpler scheme. For any given locality difficulty will be experienced in obtaining degree hour data over a period of at least 10 years.

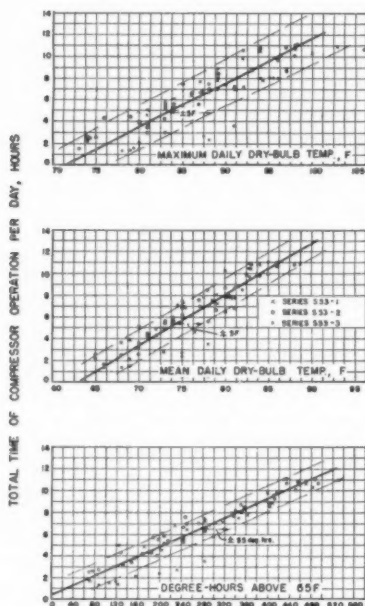


FIG. 7. COMPRESSOR OPERATION AS RELATED TO VARIOUS OUTDOOR TEMPERATURE CONDITIONS

The simplest correlation is that shown at the top of Fig. 7, in which the hours of compressor operation have been plotted against the daily maximum outdoor air temperatures. A fair correlation is obtained. Unfortunately, the daily maximum air temperature ignores the preceding history of the weather. In other words, the load on the plant will be affected by the air temperatures preceding the time when the maximum is attained.

In the middle set of curves shown in Fig. 7, the mean daily temperature has been selected as the criterion of weather and the correlation with compressor operation is shown to be better than that shown in the upper set of curves. In this case, the mean daily temperature is based on the numerical average of the maximum outdoor temperature and the minimum outdoor temperature during the preceding morning hours only. In this respect, the mean daily temperature as

commonly given by the Weather Bureau is slightly at variance since the minimum temperature is not always restricted to the hours preceding the occurrence of the maximum temperature.

From a practical standpoint, therefore, the mean daily temperature (or its equivalent, which is the degree days above some base temperature) is probably the best compromise index for use in determining hours of compressor operation. The results will not be as close as those based on degree hours above 65 F, but the advantage of easy determination of the index will more than offset the loss in accuracy. Furthermore, since the variation in degree days from one season to the next is extremely large, precision in method is not as necessary as simplicity.

The results reported in a study of the operating costs of 11 homes<sup>7</sup> indicated that the temperature basis for defining the cooling degree day should be 70 F. However, the results of the studies conducted in Research Residence No. 2 indicated that the temperature base should be approximately 65 F. In both the studies of Gilman and in Research Residence No. 2, the indoor-air temperatures were approximately the same, but in the Residence No. 2 studies the house was not occupied. It would be expected that occupancy would have the effect of lowering the datum temperature, and yet the results indicated that the datum for Residence No. 2 was 65 F despite its unoccupied condition. As was necessary in the establishment of the base temperature for winter heating, it is probable that a large number of installations should be studied before a generally acceptable base temperature for summer cooling can be determined. The possibility exists that the base temperature for one-story houses will be lower than that for two-story houses, and will be different for houses subjected to night-air cooling, and that a compromise value will eventually be necessary for practical applications.

It should be realized that the curves in Fig. 7 are based upon a ratio of calculated heat gain to rated cooling unit capacity of 1.15. Lower values of this ratio, which would mean a larger cooling unit capacity, would cause the curves to have smaller slopes, indicating fewer hours of compressor operation. It is not anticipated, however, that changing this ratio would affect the datum mean daily temperature for the Residence.

The calculations in Table 3 were made from the weather records (between May 1 and September 30, 1953) and from the average curve shown in the middle set of curves in Fig. 6. The hours of compressor operation for this 153-day period were estimated as 727 hours. Based upon power consumptions of 2 kw for the compressor, 0.32 kw for the blower, and water consumption rates of 148 gph, the estimated total consumptions were 1454 kw hr for the compressor motor, 233 kw hr for the blower motor, and 107,500 gal of water.

#### AIR DISTRIBUTION IN LIVING ROOM

Studies of systems using perimeter introduction of air through floor registers<sup>8</sup> show that room-air velocities remain below 32 fpm except immediately above the supply outlets. Other investigators<sup>9</sup> have found that it is difficult to eliminate the gravitational effects of the air when high sidewall registers designed for heating are used for cooling.

During Series S53-2, a study of the air pattern leaving the living room register was conducted. The 14-in.  $\times$  6-in. double-deflection register was located 78 in. above the floor in approximately the center of the north living room wall. The vertical vanes in the register face were set for a 22 deg horizontal deflection of

each jet. The rear vanes were set for slight downward deflection. This setting would normally be used for heating so that the warm, low density air would not impinge upon the ceiling surface.

The center of each jet issuing from the register was located both when the compressor was operating (supply-air temperatures between 61.5 and 64.0 F) and when the compressor was not operating, (supply-air temperature approximately equal to room-air temperature). In each case the average air velocity at the register face was approximately 500 fpm.

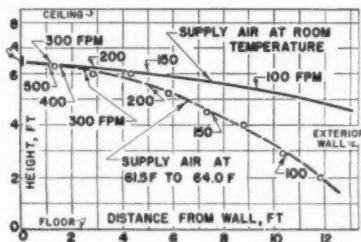


FIG. 8. VELOCITY PATTERN OF JET FROM WEST SIDE OF LIVING ROOM REGISTER, WITH 14-IN.  $\times$  6-IN. SIDE-WALL REGISTER AT POINT A

The elevations of the jets from the west side of the register are shown in Fig. 8. The supply air temperature had a decided effect upon the height of the jet as it moved across the room. When the supply air was at room-air temperature, the center of the jet was about 5 ft above the floor at a distance of 12 ft from the wall in which the register was located (1 ft 4 in. from the exterior wall). When the compressor was operating and the supply-air temperature was lower than room-air temperature, however, the center of the jet was only 2 ft above the floor at the same location. In order to reduce the drop of air into the living zone, it would be necessary to use upward deflection settings on high sidewall registers when used for summer cooling.

The temperature of the supply air had very little effect upon the horizontal throw from the register. In each case, the plan view of the jets was a straight line extending out toward the corners of the room until the velocity was reduced below 100 fpm. At this point the jet turned toward the exterior wall of the room.

### CONCLUSIONS

The air conditioning system discussed in this paper consisted of a year 'round air conditioner which contained a 2-hp mechanical condensing unit and a duct system designed only for winter heating. Registers were located in the high sidewall location on the interior walls of the rooms. The total air-flow rate was approximately 800 cfm. Studies were conducted both with continuous blower operation and with the blower cycled with the compressor.

Actual capacity of the cooling unit was found to be approximately 21,700 Btu per hr, an increase of 7 percent over that experienced with an air-flow rate of 600

cfm in an earlier investigation. The pressure loss in the duct system was 0.40 in. water column, approximately twice that experienced with the winter air-flow rate of 340 cfm. The air delivered from the high sidewall register into the living room was found to drop into the living zone. This was more pronounced when the compressor was operating and the supply-air temperature was low than it was when the compressor was not operating. The register was adjusted for winter heating and gave a slight downward deflection to the air.

Cycling the blower with the compressor produced lower room-air humidities than did continuous blower operation. Even during periods of low outdoor humidity, the amount of water condensed from the air circulated through the cooling unit was greater with cyclic blower operation than it was in more humid weather with continuous blower operation. Although cycling the blower with the compressor would seem to be desirable from the standpoint of lower indoor humidity, the possibility of discomfort due to lack of air circulation when the blower is not operating must be considered.

Great variation of the climatic factors which affect cooling unit operation can be expected from one season to another. The number of days which have a maximum outdoor temperature equaling or exceeding a given value during one season may be 100 percent greater than the number of days during which the same condition is equalled or exceeded in the following season. The variation in weather conditions is much greater for cooling than for heating. Although it is important that some relatively simple index of weather conditions which would serve as a basis for predicting compressor operation be determined, the index should be determined statistically so that predicted operating times will not be exceeded for at least three years out of every four.

Excellent correlation was found between hours of compressor operation per day and the number of degree hours above 65 F. From a practical standpoint, however, this criterion is not as good an index as degree days based on mean daily temperature. In addition, there was good correlation between hours of operation and mean daily temperature. The datum mean daily temperature above which compressor operation occurred was 65 F. The base temperature for cooling degree days must be determined from investigation of many residences, however, and it may vary depending upon the locality and type of residence.

#### ACKNOWLEDGMENTS

This paper reports one phase of a comprehensive investigation conducted in Warm Air Heating Research Residence No. 2 under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the *Engineering Experiment Station of the University of Illinois*. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation.

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## DISCUSSION

C. H. NEIMAN, Jr., York, Pa., (WRITTEN): I am extremely grateful for the opportunity to discuss this paper—not so much from the standpoint of my having any great knowledge and experience on the subject, but from the standpoint of having an opportunity to express my appreciation to those who are working so diligently on these problems which can increase our knowledge of the best in application methods to be used in residential air conditioning.

As a representative of a manufacturer of residential heating equipment, I know, first hand, the value of investigations performed by Professor Konzo and his associates. These men have done a great deal to advance the art of applying home heating equipment to give us comfort in all kinds of weather. These investigations always result in some information of practical value. I am sure that Professor Konzo and his associates will provide the same basic knowledge of home cooling as the years progress.

These remarks are not intended to detract from the work of others in the field. A great deal of good work has and is being done by others and we all know that a tremendous field of investigation lies ahead in residential cooling which will require many long and tedious hours in laboratories all over the country before we reach maturity in home cooling applications.

I find it impossible to think about the paper presented without immediately relating it to the broader aspects of the problem. I see in it certain guide posts which help to chart the course of more intelligent application of residential cooling equipment.

Those of us who are associated with both home heating and home cooling know what I mean when I say that home cooling requires some special considerations as compared to commercial cooling. We are quick to apply the cooling equipment using the same considerations as we have in commercial installations only to find that we have over-engineered the job. The cost is either too high, the equipment too large, or there is no space for the cooling tower, or no good way to dispose of the condenser water.

The potential in home cooling is large, but in my humble opinion will only be available to us when we can find the way to provide comfort at a cost that the majority of home owners can afford to pay.

You may wonder what connection the foregoing remarks have with the paper presented, but I see in the paper several things which add emphasis to these remarks.

Many have talked about the proper sizing of home cooling equipment, and have recommended slightly undersizing the equipment rather than oversizing for peak loads.

This paper presents some facts which in my mind confirm the wisdom of smaller size equipment.

The authors have shown that better humidity control is obtained when the blower and the compressor cycle together. However, there is a question whether the discomfort due to lack of air circulation might not overshadow the comfort gained due to better humidity control. Therefore, the logical conclusion would be to operate both the compressor and blower continuously or as nearly continuously as possible.

One way to accomplish this type of operation is to size the equipment in such a manner that it will just carry the load.

A recommendation such as this without some supporting information would be useless and, therefore, I would like to give you some facts obtained during the 1953 and 1954 cooling season which lead us to the belief that comfort can be obtained from equipment which has a capacity lower than the calculated heat gain.

Mention has been made in the paper of the method of determining heat gain outlined in Manual 11 of the *National Warm Air Heating and Air Conditioning Association*.

We find that this method results in a total heat gain greater than some other methods. However, we have never found a method of calculating heat gain which matches the apparent actual heat gain as indicated by the comfort satisfaction of the occupants. Therefore, to determine the actual heat gain by calculation is a difficult task and to the best of my knowledge no one can say that one particular method is correct. It is reassuring, however, to know that a great deal of work is being done on this subject.

Reference 1 of the paper contains data which supports the evidence that heat gain calculations exceed the real heat gain.

We have installations where the occupants report satisfaction with 1½ hp air-cooled units and the heat gain calculations indicate that a 2 hp or larger unit is required. These homes are approximately 1000 square feet in floor area. Temperature readings show the inside dry bulb being maintained at 80 to 85 F with 106 F dry bulb outside.

Obviously, I am not suggesting a movement to undersize all applications but we have had sufficient successful experiences to indicate that further research should be done to firm up the proper sizing of cooling equipment to provide satisfactory comfort, and keep first cost to a minimum.

The paper also points up another very important consideration. The need for a method to be used in estimating cost of operation is apparent especially when the smaller, medium and lower-priced homes are to be considered for air conditioning.

It would appear that from the data presented, mean daily temperature is a good basis on which to estimate unit operation. This information is generally available and reliable.

The paper, however, is not clear on the method used to convert degree days to hours of compressor operation, and the authors should explain this in more detail.

I would assume that some factor derived from experience would apply to each type of installation. This factor might take the form of degree days per hour of compressor operation.

If such a factor were used in the calculations summarized in Table 3, the degree days per hour of compressor operation would be

$$\frac{1272}{727} = 1.75$$

This factor would correspond to the degree days per gallon used in predicting fuel oil deliveries for heating. However, each type of system requires a different factor and the accuracy of the prediction depends entirely on the accuracy of the factor.

However, no matter what method is used it would seem that the conversion of degree

days for cooling to hours of compressor operation would depend on the type of installation, and how it is used.

The authors indicate in their summary that additional facts must be obtained from many residential installations, and we will look with interest on future reports of their progress. They are making a real contribution to the general knowledge of home cooling and should be complimented and encouraged to continue.

S. A. HEIDER, Washington, D. C., (WRITTEN): This paper illustrates very effectively how the severity of summer weather varies from season to season. Table 2 shows there were 50 days of 90 F or over in Urbana, Illinois, during 1936 as against only 19 days in 1935. With such great variations, temperature averages are rather poor yardsticks of weather conditions.

In addition to this shortcoming, averages don't give the designer the kind of information he needs, namely, a choice of temperatures and some kind of comparison between temperatures. More and more designers are becoming interested in data giving the *incidence of occurrence* of a series of temperatures, i.e., the number of times that selected temperatures were equaled or exceeded over a given period of record. This gives the designer a choice of temperatures to suit his particular needs, and also enables him to tell his client just how many days out of the season the cooling system can be expected to meet the load.

The paper also points out that a single over-all load index cannot successfully take into account all the variables of compressor operation, including such items as internal load or night-time precooling by ventilation. However, the paper does suggest that a single, relatively simple index of weather conditions would be very helpful.

Charts are presented in the paper which plot compressor operation during tests against maximum daily dry bulb, minimum daily dry bulb, and degree-hours above 65 F. Dry bulb was probably used because it is the commonest and most important single weather element. However, a dry bulb index does not take into account fresh air load, solar radiation, and wind. Fresh air load was not mentioned for this residence, but is important for most other buildings.

Solar radiation is a considerable item but not nearly so important as fresh air load. Wind is a rather minor factor. The fresh air or moisture infiltration load is frequently a considerable load and is usually measured by wet bulb. Wet bulb, in my opinion, ought therefore to be added to the dry bulb as an additional factor obtained from the same simultaneous readings of record.

To simplify the resulting dual index, consideration might be given to using a combined dry bulb and wet bulb index as a single figure. Such a *combined outdoor temperature* has already been proposed by a group of Federal agency engineers for use in determining the need of one locality against another for cooling equipment in Federal buildings. This combined outdoor temperature would be, in my opinion, a better base for degree-hours of compressor operation because it contains the important element of moisture load.

K. T. DAVIS, Cleveland, Ohio, (WRITTEN): In considering the paper just presented, it should be kept in mind that this and earlier summer cooling studies made in Research Residence No. 2, were aimed at determining what and how much should be done to a well designed distribution system originally intended for heating in order to obtain a good summer conditioning job. Air direction and diffusion from the registers, capacity of the duct system, performance requirements of the fan, and the variation of cooling capacity caused by possible reduction in air flow rate are therefore points for consideration. It is believed in this and the previous 1953 report, the authors have contributed information which has significant practical value in the conversion of existing warm air heating systems into all-year comfort producing plants.

In connection with operating costs, there are many questions which might be asked. Assuming a constant inlet temperature, how much did the water flow rate vary under different evaporator loads? How much did the electrical input to the compressor vary

under similar conditions? Actually, however, these factors appear to be of minor influence in forecasting cost of operation as compared with our own inability either to accurately forecast weather conditions or anticipate the indoor temperature which the home owner will maintain. It is interesting to note good correlation of hours of compressor operation with outdoor temperatures can be obtained *after* it has happened. But annual variations in summer conditions may be so great that most predictions of operating costs probably should be qualified on a *cents per degree day* or cents per degree hour basis, and such values then applied to local weather experience with logical explanation of possible deviations.

Perhaps the authors could offer a little added information on two points raised in the paper. First, is it correct to assume the humidity ratios shown in Fig. 4 were obtained with instruments in the rooms rather than in the return duct as in Fig. 5? Second, would the authors further explain the reasons for the 340 cfm and 565 cfm air flow rates for heating, and rationalize their findings that at an 800 cfm flow rate for cooling the pressure losses in the system "are approximately double those used in the original design of the duct system" for heating?

G. V. PARMELEE, Cleveland, Ohio, (WRITTEN): The authors have presented much useful additional information on the performance of residential cooling systems and it is hoped that these studies will continue. Of particular interest is the correlation of hours of compressor operation with outdoor weather conditions. The authors have commented on the difficulty of using a single index, because four separate factors affect the cooling load namely, outdoor dry-bulb and wet-bulb temperatures, solar radiation, internal heat and moisture sources, and wind speed and direction. I wish to suggest that the authors try to make a more detailed analysis of the relationship between compressor operation and weather. The following procedure might be used. Total solar radiation received per day on a horizontal plane should be a fairly accurate index of the total effect of this component. Either maximum daily dry-bulb or mean dry-bulb might be an adequate index of this component, while mean or maximum wet-bulb temperature would indicate the magnitude of the heat gain due to infiltrating sensible and latent heat. If arbitrary ranges in the values of each of these factors were set up, hours of compressor operation could be classified for simultaneous values of these weather factors. If there are sufficient data it would be possible to plot hours of operation against one variable with the other two held constant. Inclusion of the fourth element, wind velocity, might not be necessary, if it is found that day-to-day variations in total wind movement are small.

The practical need for a good way of estimating operating costs is recognized as being urgent and the method suggested by the authors may be suitable for interim use. However, the difference between their datum point of 65 F, and the datum of 70 F suggested by Gilman, indicates the need for a more fundamental analysis of the problem before any extensive and costly attempt is made to develop cooling degree days.

The correlation method suggested requires considerable test data for a variety of combinations of the weather factors. It also has the disadvantage that it ignores the carry-over of stored heat and moisture from one day to the next. This could be important if weather conditions on successive days differed greatly. However, it might shed additional light on the subject and lead to some more suitable basis of correlation.

Another method of attack would be to impose different weather conditions, arbitrarily selected, on a thermal circuit for a given structure, and work out the cooling loads. The advantages of this procedure are very great. Nevertheless, very careful field investigations, of the type carried on by the authors, are an essential corollary to such a procedure.

L. A. HALL AND S. F. GILMAN, Syracuse, N. Y., (WRITTEN): The testing of an actual residence is a most difficult task and inevitably leads to analyzing the type of data

typified by Fig. 4, where the scatter of points is considerable. Since there are always several variables which cannot be controlled, such papers as these provoke many questions and comments.

It would hardly seem necessary to investigate the effect of increasing the air flow rate from 600 to 800 cfm because such information should be obtainable directly from the equipment manufacturer. Presuming that there was some reason other than obtaining the capacity, what notable differences or similarities were there between the two series of studies at the two air flow rates? It is noted that heat flow studies were made during the previous season and that this paper reports heat meters were installed during these tests; however, no further reference to heat meters is found. Again it is wondered if comparisons could be made between the two different test conditions.

It does not seem that the effect of intermittent and continuous fan operation, as well as addition of moisture, can be properly analyzed without taking into account the performance characteristics of the direct expansion equipment used. For example, the statement that the release of 4.5 pounds of moisture resulted in a larger amount of condensate is to be expected because the sensible heat factor of the equipment is a function of the entering wet bulb temperature. However, when and how long the compressor operates depends to a great extent on the sensible heat loading of the structure. The discussion points out, and Fig. 5 presumably shows, that the weight of condensate collected is not the same for intermittent and continuous fan operation for nearly identical days. It should be noted, however, that although the days may be somewhat similar with respect to outdoor wet and dry bulb, the compressor loadings are far from similar. In the top portion of the figure the compressor operates only about half the time, whereas in the lower portion it operates about three-quarters of the time. This in itself would lead to more condensate being collected for the latter case.

It is stated that different thermostat settings will affect the number of hours of compressor operation, which is certainly logical. However, the reason for this is stated to be that the "rate of rise of room air temperature in a residence may be as low as  $\frac{1}{2}$  to 1 degree per hour without cooling." This is very confusing. How is the number of compressor operating hours a function of conditions that occur in a residence that is *not* air conditioned? The statement is also made that the rooms in a one-story house will require more cooling and at an earlier hour than the first story rooms of a multi-story residence. How this ties into the problem is difficult to see. To predict operating costs all one has to do is evaluate the *total number of Btu's* which must be extracted from a given residence for the entire cooling season. Construction differences should be accounted for in the evaluation of the cooling load. This cooling load, whether computed for a design day or for each day of the year, must enter the correlation equation for predicting operating costs.

It is interesting to note that, in spite of the fact that the calculated heat gain is 15 percent larger than the equipment capacity, the unit operated a maximum of 11 hours per day even when the design outdoor dry bulb was exceeded 14 times. It appears that the heat gain calculation procedure gives conservative results. Our experience in field testing of residences throughout the country indicates the capacity of this unit could be reduced to about 12,000 Btu per hr and satisfactorily cool the residence with a maximum temperature swing of four degrees.

In discussing the air distribution in the living room, the statement that room air velocities remain below 32 feet per minute with perimeter introduction of air is a broad generalization and should be qualified. Finally, the pressure loss at 800 cfm is given as 0.40 in. of water in the system and it is stated that this value is approximately twice that with the winter air flow rate of 340 cfm. Since for a constant system the pressure will vary as the square of the cfm, it should be more like 5 times as much unless the system resistance has been altered by dampers. Was this what happened?

It appears that more valid and useful results could be obtained from tests such as these if the analyses were made on an hour-by-hour basis instead of a seasonal basis.

Information such as presented is of some value because it shows general performance characteristics. However, it offers nothing that can be extended and generalized for formulating specific engineering design information. This residence is highly instrumented and therefore ideally suited for more fundamental and basic approaches to the mechanism of periodic heat transfer phenomena. It is hoped that the facilities and research efforts will eventually be directed toward areas where important contributions of a more basic nature can be made.

**AUTHORS' CLOSURE (H. T. GILKEY):** Mr. Davis asked two specific questions concerning the cooling water flow rate and the power consumption of the cooling unit. The average water flow rate was 149 gal. per hr of compressor operation, or approximately 2.5 gpm. An inspection of the orifice meter recorder charts indicates that the flow rate was a maximum at the start of each operating cycle and decreased approximately 0.1 gpm during a 15 or 20 min cycle. It should be emphasized that water conserving devices were not used with this installation. The average power consumption was approximately 2.0 kw and did not appear to change appreciably with either evaporator load or cycle length.

The humidity ratios shown in Fig. 4 were obtained with instruments located in the return air plenum. These instruments were the same as those used to obtain the information shown in Fig. 5. The data in Fig. 4 are based upon 24 hr averages of dry and wet bulb temperatures.

The heating air-flow rates were 340 cfm in Research Residence No. 2 when only the upstairs was heated and 560 cfm when both the basement and the upstairs were heated. The duct system was designed for the higher of the two flow rates, 560 cfm, and the extended plenums from the unit were sized for this condition. The pressure drop experienced when circulating 800 cfm was approximately twice that experienced with 560 cfm. It is interesting to note that 560 multiplied by the square root of 2 is approximately equal to 800.

Both Mr. Davis and Mr. Neiman commented on the need for some method of getting an index of operating cost. It was suggested that this could be based on an average cost per cooling degree day. Of course, this must be the objective, but it must be the result of work by many investigators because of the large number of variables involved. These include weather variables, house construction variables, occupancy variables, and equipment variables. The data shown in this paper are based upon experimental results and no basis of prediction has been suggested.

Mr. Heider commented that no fresh air was introduced into the Residence. He also mentioned that it is commonly needed in other types of construction and that in some areas it is common practice in residential cooling. Studies have been conducted both with and without the introduction of ventilation air into Residence No. 2, and, at least for the unoccupied Residence, the primary effect of ventilation air was found to be increased operating times.

It is pointed out by Messrs. Hall and Gilman that it is difficult to make detailed analyses of continuous and cyclic blower operation when direct expansion equipment is used, and that in Fig. 5 the compressor loadings are quite different for the two days shown. This is due in part to differences in weather conditions, an inevitable situation despite attempts to choose days of maximum similarity. It must also be considered that the differences in loading may be in part the result of the type of blower operation and as such can be considered typical.

Attention is also called by Messrs. Hall and Gilman to the absence of information on periodic heat flows, and temperature profiles. It should have been mentioned in the paper that the results of these studies were the same as those reported in Reference 1. In addition, References 1 and 8 indicate there might be some difficulty in cooling the Residence with a unit having a capacity of 12,000 Btuh.

Each of the discussors suggested the possibility of including factors in addition to dry-bulb temperature in attempting to predict operating times. Mr. Parmelee suggested dry-bulb temperature, wet-bulb temperature, and total solar radiation, and he

suggested a method whereby the effect of each might be determined. Similar procedures were tried on the data obtained in connection with these studies and little or no correlation was obtained. Further attempts along these or similar lines must be made, for we must be able eventually to isolate the factors which contribute to heat gain. The immediate need, unfortunate though it may be, is to determine the overall effects of the various weather, equipment, and occupancy factors on conditions within a house. This leaves us in the position of having to learn a lot about the forest without having an opportunity to study the individual trees.

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## OKLAHOMA REGIONAL MEETING, 1955

OKLAHOMA CITY, OKLAHOMA

The Regional Meeting at Oklahoma City, April 14-16, 1955—the first held under the new ASHAE name—broke the attendance record with a total registration of 241. This was made up of 123 Members, 44 Guests and 74 Ladies. The largest attendance was from Oklahoma but registrants were present from 17 states and the District of Columbia and Canada.

Pres. John E. Haines opened the morning session at 10:00 a.m., on Saturday, April 16th, in the Crystal Room of the Skirvin Hotel, and at once called on W. J. Collins, Jr., general chairman, who extended the welcome of the Oklahoma Chapter, outlined the features of the program, and asked for all to register.

President Haines told of his recent visit to the Canadian Conference in Toronto, and mentioned that he had visited more than 40 of the present 59 chapters in the last few years. He also expressed the hope that Regional Meetings would continue and that the Society and its research activities would continue to meet its responsibilities to the industry and the general public.

He reported also that the organization of two new chapters at Baton Rouge and Springfield, Mass. is approved. For the first time the total membership of the Society, with present applications, exceeds 10,000. He was also much pleased with the activities and accomplishments of the various committees so far this year. He took the opportunity of outlining some of the problems currently facing the Officers and Council and indicated that solutions are being actively sought.

President Haines then turned the meeting over to Second Vice Pres. P. B. Gordon, and 2 technical papers were presented and discussed (see Program on p. 279). The first of these papers had to do with pressure losses in divided-flow fittings, while the second presented the results of a study on panel cooling for a residence.

At the afternoon session, opened by President Haines at 1:30 p.m., Saturday, April 16th, E. R. Kaiser, Director of Research, outlined the nature of the 23 projects now making up the ASHAE program. Nine are at the Laboratory and 14 are at cooperating institutions. He explained how they fit into the needs of a project for design and operational information. He stressed objectives rather than methods, and made it clear that in many cases the objectives cannot be reached without long continued study. The needs for research data are increasing, problems are pressing

for solutions, and he asked patience since he believed results will produce returns with a value at least double the costs.

President Haines then introduced B. H. Jennings, who presided during the presentation and discussion of 2 technical papers. One of these papers dealt with the subject of an analysis method to predict the behavior of solid adsorbents; while the second, a result of a cooperative study between the Laboratory and Tulane University, reported an analysis of weather data for New Orleans.

President Haines resumed the chair and called upon B. H. Spurlock, Jr., to present the Report of the Committee on Resolutions which follows:

#### RESOLUTIONS

WHEREAS the first Regional Meeting of the Society under its new name ASHAE is about to be made a part of the history of the Society, and

WHEREAS the meeting was held in Oklahoma City, capital of the Sooner state, and whose capital grounds have oil wells instead of putting greens, and located in the vicinity of the home of the world's only Will Rogers, and

WHEREAS, under the excellent direction of our Officers and Staff the aims of the Society have been much advanced through the presentation of technical papers, and

WHEREAS, the Oklahoma Chapter, through its Committee on Arrangements of which W. J. Collins, Jr., its general chairman, have provided a most entertaining program and have been solicitous of the welfare and comforts of its guests; therefore

BE IT RESOLVED that we express our sincere gratitude and thanks:

TO the Oklahoma Chapter, its officers and committees, who have made this an exceptional meeting,

TO W. J. Collins, Jr., general chairman and G. E. Ervin chapter president, for their successful effort in conducting this meeting,

TO the hotel management who has been thoughtful of our needs and comfort,

TO the authors of the technical papers and those who participated in the discussions,

TO the ladies of Oklahoma Chapter for their wonderful hospitality to the visiting ladies,

TO Tom Collins for his humor and wit,

TO President John E. Haines, the officers and committees, and to all others who have contributed their time and efforts to insure a successful meeting.

Respectfully submitted,

The Resolutions Committee

B. H. Spurlock, Jr., *Chairman*,  
Boulder, Colo.

John Everetts, Jr., Philadelphia, Pa.

W. A. Grant, Syracuse, N. Y.

The resolutions were adopted unanimously and the meeting was adjourned.

## PROGRAM REGIONAL MEETING

Skirvin Hotel, Oklahoma City, Oklahoma—April 14-16, 1955

## Thursday—April 14

- 10:00 a.m. Executive Committee (*10th Floor Studio*)  
1:30 p.m. Finance Committee (*10th Floor Studio*)  
3:00 p.m. Chapter Relations Committee (*10th Floor Studio*)

## Friday—April 15

- 9:30 a.m. Council Meeting (*Regency Room*)  
9:30 a.m. Long Range Research Planning Committee (*Blue Room*)  
10:30 a.m. Informal Golf at Lincoln Park  
11:00 a.m. REGISTRATION (*Mezzanine*)  
1:30 p.m. Sightseeing: Oklahoma City; TV Studio  
2:00 p.m. Research Executive Committee (*Blue Room*)  
4:00 p.m. Informal Conference of Chapter Members (*East Room*)  
6:30 p.m. Oklahoma Roundup: (*Balinese and Crystal Rooms*)  
Social Hour, Buffet Supper, Entertainment

## Saturday—April 16

- 9:00 a.m. REGISTRATION (*Mezzanine*)  
10:00 a.m. TECHNICAL SESSION (*Crystal Room*)  
Call to Order: President John E. Haines  
Report of the President: John E. Haines  
P. B. Gordon, *Chairman*  
Pressure Losses of Divided-Flow Fittings, by S. F. Gilman, Syracuse, N. Y., presented by Mr. W. A. Grant,  
Panel Cooling for a Residence, by R. R. Irwin, Stillwater, Okla., and presented by Prof. Irwin.  
12:30 p.m. LADIES LUNCHEON AND FASHION SHOW (*Continental Room*)  
1:30 p.m. TECHNICAL SESSION (*Crystal Room*)  
B. H. Jennings, *Chairman*  
What's New in ASHAE Research: E. R. Kaiser, Director of Research.  
Weather Data Analysis for Cooling System Design, by G. V. Parmelee, G. E. Sullivan and A. N. Cerny, Cleveland, Ohio, presented by Mr. Parmelee.  
An Analysis Method for Predicting Behavior of Solid Adsorbents in Solid Sorption Dehumidifiers, by W. L. Ross and E. R. McLaughlin, University Park, Penna., presented by Prof. McLaughlin.  
7:00 p.m. BANQUET (*Persian Room*)  
Toastmaster: Earle W. Gray  
Invocation: Rev. Thomas C. Davies, Westminster Presbyterian Church  
Indian Ceremony: Chief Jasper Sawneah, Kiowa Tribe  
Speaker: Tom Collins, Kansas City, Mo.  
Subject: Two and Two Aren't Always Four

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## PRESSURE LOSSES OF DIVIDED-FLOW FITTINGS

By S. F. GILMAN\*, SYRACUSE, N. Y.

AIR DUCT systems are comprised of sections of straight pipe joined by fittings which can be classified as either the through-flow or the divided-flow type. The abrupt contraction illustrated in Fig. 1 is typical of through-flow fittings. For engineering purposes the loss of head in this fitting is expressed by the Carnot-Borda equation

$$H = (V_1 - V_2)^2 / 2g \quad (1)$$

in which

$H$  = loss of head, feet of fluid.

$V$  = mean velocity at cross section designated by subscript, feet per second.

$g$  = acceleration due to gravity, feet per (second) (second).

This equation generally appears in the form

$$H = [(1/C_c) - 1]^2 V_2^2 / 2g \quad (2)$$

where  $C_c$  is a coefficient of contraction which depends on the ratio of the downstream and upstream areas; i.e.  $A_2/A_1$ . For air conditioning systems the volume rate of flow of air in each portion of the duct system is usually known in advance. Therefore, once the size and shape of a through-flow fitting is selected, the pressure loss can be evaluated from engineering data available in the literature, together with the continuity equation for constant density,

$$Q = AV \quad (3)$$

in which

$Q$  = volume rate of flow.

$A$  = cross section area.

A typical divided-flow fitting is illustrated in Fig. 2 and in contrast with the through-flow type, the continuity equation involves three cross sections; thus

$$Q_u = Q_b + Q_d \quad (4)$$

As indicated by the broken line in Fig. 2, a streamline can be envisioned as separating the flow that is diverted into the branch,  $Q_b$ , from the straight-through

\* Head, Air Conditioning Systems and Equipment Section, Research Department, Carrier Corp. Member of ASHAE.

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flow,  $Q_d$ . The shape of such a streamline and its point of contact with section  $u$ , over which the velocity is constant, will vary with the proportion of the total flow  $Q_u$  diverted into the branch. Since the flows on either side of the streamline have different histories, the mechanical energy per unit mass at section  $b$  will not necessarily be equal to the corresponding quantity at section  $d$ . Consequently, it is necessary to consider the energy dissipation in the diverted and straight-through portions separately.

Another distinguishing characteristic of divided-flow fittings is that the pressure loss is a function of the ratio of *two* characteristic velocities or flow rates; consequently, compared with the through-flow fitting the divided-flow type involves one additional variable. This was demonstrated experimentally by Vogel<sup>1</sup> and by Holl<sup>2</sup> and Konzo<sup>3</sup> through dimensional analysis as well as experiments.

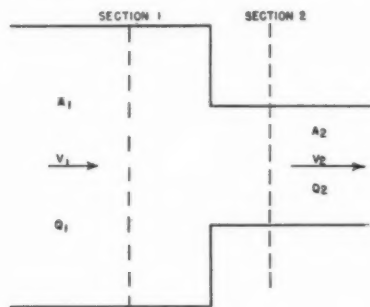


FIG. 1. ABRUPT CONTRACTION, A TYPICAL THROUGH-FLOW FITTING

Considerable research on divided-flow elements has been conducted by Thoma and his associates at the *Munich Hydraulic Institute*. Water was used as the fluid and the investigations were restricted to elements having a main pipe of the same size and shape at cross sections  $u$  and  $d$  in Fig. 2. Vogel<sup>1,4</sup> reported an investigation of various right-angle pipe tees; Petermann<sup>5</sup> and Kinne<sup>6</sup> reported studies of 45 and 60 deg oblique-angled pipe branches, respectively. Other reports of experiments utilizing water are those concerning the Boulder Dam<sup>7</sup> and a power station in Switzerland<sup>8</sup>. McNown and Hsu<sup>9</sup> have presented a theoretical treatment in which conformal mapping is utilized to determine the characteristics of divided flows. Concerning types of divided-flow fittings common to air conditioning, laboratory investigations have been reported by Korst<sup>10</sup>, Holl<sup>2</sup> and Konzo<sup>3</sup>, as well as in a series of 12 reports issued by the U. S. Navy<sup>11</sup>.

Although considerable research has been conducted, the behavior of the pressure loss curves has yet to be suitably explained. Also, the results of the several investigators often appear to be in poor agreement. Although this poor agreement is partially due to differences in experimental results, it is also partially due to the different bases used for expressing the results. Sufficient information now appears available so that the behavior of the loss curves for the diverted-portion of the total flow can be explained in a rational and conclusive manner. In addition, the probable best relation expressing the pressure loss in the straight-through section can be

indicated, although the available data are not nearly as extensive. Since the two flows in the divided-flow fitting must be considered separately, that in the straight through section is considered first.

#### STRAIGHT-THROUGH SECTION

Of the two flows, that in the straight-through section is the less complex because in continuing down the main (see Fig. 2) it does not experience the change in direction that the flow into the branch does. Moreover, the area at section d is ordinarily equal to that at section u or somewhat less, but rarely is the reduction so great that  $V_d$  exceeds  $V_u$ . Hence the flow in the main is almost invariably a retarded flow, whereas the flow in the branch can be either retarded or accelerated with

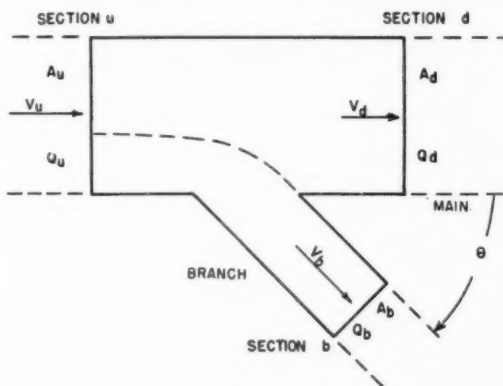


FIG. 2. TYPICAL DIVIDED-FLOW FITTING

respect to  $V_u$  depending upon the proportion of the total flow at section u diverted into the branch.

The usual expression for the loss of head in the straight-through section is

$$H_d = [P_u + (V_u^2/2g)] - [P_d + (V_d^2/2g)] - {}_uF_d \quad (5)$$

in which

$P$  = static head at measuring station at designated cross section, feet of fluid.  
 ${}_uF_d$  = nominal loss of head due to friction in the pipe section between sections u and d, feet of fluid.

It should be noted in Equation 5 that the nominal friction loss  ${}_uF_d$  is deducted from the difference in total heads; hence,  $H_d$  is considered to be concentrated at the section at which the main and branch pipes join. The head loss is thereby expressed on the conventional *no length* basis, which has the advantage of permitting direct comparison of results by different investigators whose measuring stations (sections u and d in Fig. 2) were located at different distances from the junction.

Most of the investigators expressed results as dimensionless plots of either the flow-rate ratio or velocity ratio as independent variable and a *loss coefficient* as the

dependent variable. This loss coefficient is defined sometimes<sup>11</sup> as

$$\lambda_d = H_d/(V_d^2/2g) \quad \dots \dots \dots (6)$$

and sometimes<sup>1</sup> as

$$\lambda'_d = H_d/(V_u^2/2g) \quad \dots \dots \dots (7)$$

Hence, care must be exercised in interpreting results as the loss may arbitrarily be referred to either the upstream or downstream velocity head.

Comparison of the reports of Vogel<sup>11,14</sup>, Petermann<sup>5</sup> and Kinne<sup>6</sup> shows that a single curve can be used to closely represent their results. This means that the loss of head in the straight-through section is independent of the size of the branch and its angle with respect to the main. Such a curve is given by the parabola

$$H_d/(V_u^2/2g) = 0.35 (Q_b/Q_u)^2 \quad \dots \dots \dots (8)$$

The physical significance of this equation can be made clear by expressing it in a different form. By the continuity equation (Equation 4) it is possible to express (8) as

$$H_d/(V_u^2/2g) = 0.35 [1 - (Q_d/Q_u)]^2 \quad \dots \dots \dots (9)$$

or, since the areas at sections u and d are equal,

$$H_d/(V_u^2/2g) = 0.35 [1 - (V_d/V_u)]^2 \quad \dots \dots \dots (10)$$

which simplifies to

$$H_d = 0.35 [(V_u - V_d)^2/2g] \quad \dots \dots \dots (11)$$

Comparison of this equation with the Carnot-Borda equation for abrupt expansion given by Equation 1 shows that the loss of head in the straight-through section of the divided-flow element is 0.35 of the Carnot-Borda loss for the same velocities.

Equation 11 is based on experiments using water as the fluid. For the flow of air in ducts the commonly used equation is, when expressed in general units of feet of fluid flowing,<sup>12,13</sup>

$$H_d = 0.5 [(V_u^2 - V_d^2)/2g] \quad \dots \dots \dots (12)$$

This equation is not equivalent to Equation 11 and generally yields considerably larger values of loss for similar flow conditions. Comparative values are shown in Fig. 3. It should be noted that Equation 11 is restricted to the condition  $A_u = A_d$  whereas the source<sup>12</sup> of Equation 12 imposes no restrictions on the respective areas. However, the latter equation is not valid for  $V_d > V_u$  since a negative result would be obtained. The work of Konzo<sup>3</sup> indicates the factor 0.5 in Equation 12 is very conservative for the majority of flow conditions, and that it ordinarily varies from 0.25 to as low as 0.05.

From the Navy reports<sup>11</sup> covering a wide range of branch angles for round and rectangular ducts, a representative value for the straight-through section for branching angles between 30 and 90 deg is

$$\lambda_d = 0.25 \quad \dots \dots \dots (13)$$

which again indicates the loss is essentially independent of the size and shape of the branch and its angle with the main. These reports also illustrate the caution which

<sup>1</sup> Exponent numerals refer to References.

must be exercised when interpreting results of various investigators. Detailed examination of the laboratory data shows that Equation 13 includes the nominal friction loss between measuring stations separated by approximately four feet of duct. When this loss is deducted in accordance with Equation 5, the value reduces from 0.25 to 0. Since the tests were conducted with equal velocities in the 8-in. diameter upstream and 5.5-in. diameter downstream cross sections of the fittings, the latter agrees with that which would be obtained from Equations 11 and 12.

Since the flow along the main does not experience the change in direction that the flow into the branch does, the pressure loss in the main would ordinarily be expected to be less than that in the branch. This is clearly shown by the Navy tests<sup>11</sup> con-

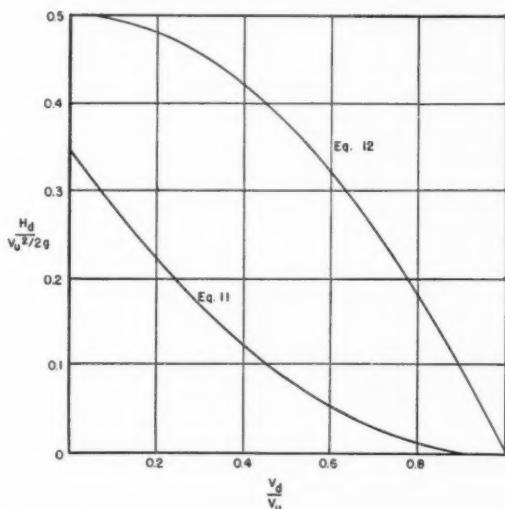


FIG. 3. COMPARATIVE RESULTS FOR STRAIGHT-THROUGH SECTION OF DIVIDED-FLOW FITTING

ducted with the fitting shape of Fig. 4. For branching angles of 90, 60, 45 and 30 deg, the straight-section loss is 14, 23, 39 and 58 percent, respectively, of the head loss in the branch. Thus, in the case of the more commonly used fitting (90 deg branch) the straight-through section is very efficient in comparison with the branch section.

From the present state of knowledge, Equation 11, which predicts the lower loss, appears to best represent the actual flow conditions in the straight-through section of divided-flow fittings.

#### DIVERTED-FLOW SECTION

As with the straight-through section, the loss of head in the diverted-flow section is a function of the flow-rate ratio  $Q_b/Q_u$  or the velocity ratio  $V_b/V_u$ . Conse-

quently, the loss in the branching section also involves one more variable than through-flow elements. Referring to Fig. 2, the ratios are related by the equation

$$Q_b/Q_a = (A_b/A_a)(V_b/V_a) \quad (14)$$

The use of velocity ratio as independent variable has been found to provide better correlation than the flow-rate ratio<sup>3,14,15</sup>. Therefore, the flow configuration in terms of the velocities in the immediate vicinity of the entrance apparently contributes most to the formation of the entrance loss.

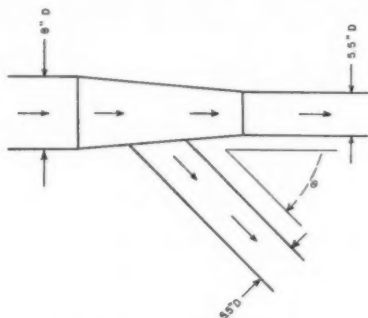


FIG. 4. FITTINGS USED IN SEVERAL TESTS BY U. S. NAVY

For the fitting illustrated in Fig. 2 the loss of head is given by

$$H_b = [P_u + (V_u^2/2g)] - [P_b + (V_b^2/2g)] - {}_uF_f - {}_bF_b \quad (15)$$

in which

${}_uF_f$  = nominal friction loss in the pipe section between  $u$  and the junction of the branch pipe, feet of fluid.

${}_bF_b$  = nominal friction loss in the pipe section between the junction of the pipe and section  $b$ , feet of fluid.

Hence, the loss is expressed on the *no length* basis.

The loss coefficient used in compiling the data in dimensionless form is sometimes<sup>2</sup> defined as

$$\lambda_b = H_b/(V_b^2/2g) \quad (16)$$

and sometimes<sup>7</sup> as

$$\lambda'_b = H_b/(V_u^2/2g) \quad (17)$$

Therefore, as with the straight-through section, care must be exercised when interpreting the results of different investigators.

Typical results for the diverted flow are presented in Fig. 5. The original data have been transposed from the Boulder Dam report<sup>7</sup> with the abscissa changed from  $Q_b/Q_a$  to  $V_b/V_u$  by application of Equation 14. Comparison of the curves shows that the conical entrance to the branch yields much lower values at the higher velocity ratios. That both curves join at the point  $V_b/V_u = 0$  and  $\lambda'_b = 1.0$  is significant because it leads to intuitive reasoning that the behavior of the curves can be explained in a rational manner.

THE END POINT  $V_b/V_u = 0$ 

Consider the fitting shown in Fig. 6(A) which has a branch oriented at some angle  $\theta$ . The given flow conditions represent the limiting case of no flow in the branch; i.e., the branch damper is completely closed. The branch then acts as a static-pressure tap connected to the main. The pressure head in the branch is

$$P_b = P_u - u F_j \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

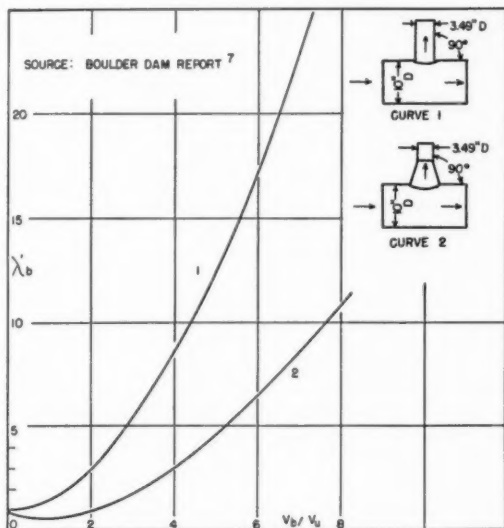


FIG. 5. LOSS COEFFICIENT ( $\lambda'_b$ ) OF THE DIVERTED FLOW FOR TWO DIVIDED-FLOW FITTINGS

and Equation 15 reduces to

$$H_b = V_u^2/2g - V_b^2/2g - j F_b \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

However, since there is no flow in the branch the last two terms disappear and we have

$$H_b = V_u^2/2g \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

and from Equation 17,

$$\lambda'_b = 1.0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

which agrees with Fig. 5 at  $V_b/V_u = 0$ , the condition for which Equation 21 applies.

That this result could be predicted was evidently not realized when the Boulder Dam report<sup>7</sup> was formulated, because at  $V_b/V_u = 0$  the empirical equations given in the report yield 0.8 and 0.5 for curves 1 and 2, respectively.

THE END POINT  $V_b/V_u = \infty$ 

Having considered the limiting case  $V_b/V_u = 0$ , attention is now turned to the other limiting case of  $V_b/V_u = \infty$  which is illustrated in Fig. 6(B). If the diameter

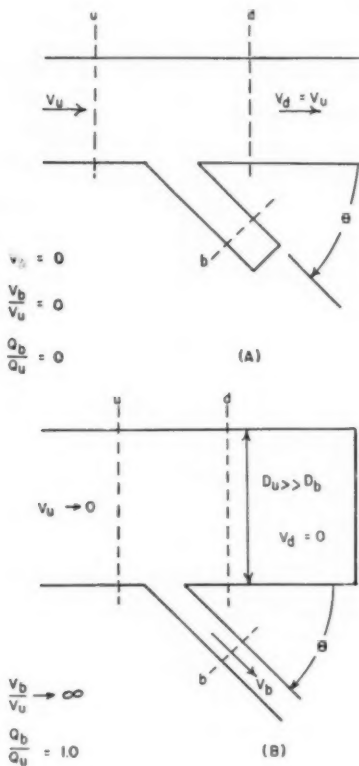


FIG. 6. LIMITING CONDITIONS FOR DIVIDED-FLOW FITTING

of the main is very large,  $V_u$  becomes, essentially, zero. The loss of head at the junction is then equivalent to the loss of head in a pipe entrance flush with the wall of a quiet reservoir. According to Weisbach<sup>16</sup> this loss is

$$H_b/(V_b^2/2g) = 0.50 + 0.30 \cos \theta + 0.23 \cos^2 \theta \quad (22)$$

in which

$\theta$  = branching angle as defined in Fig. 6.

From this equation and Equation 20 the two end points of the branch curves for fittings of the geometry shown in Fig. 6 can be determined. It now remains to explain the shape of the curves connecting these end points.

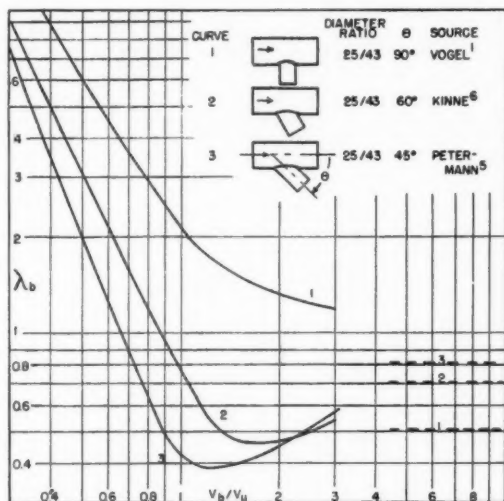


FIG. 7. INFLUENCE OF BRANCHING ANGLE  $\theta$

#### THE CONDITION OF MINIMUM LOSS

Typical curves relating velocity ratio to the loss coefficient defined by Equation 16 are presented in Fig. 7. The area ratios are the same but the branching angles are different. Since curve 1 does not have a minimum whereas the others do, and since curves 2 and 3 cross, the initial impression would be that no particular correlation could exist between them. However, it can be shown that they have a logical behavior.

First, from Equation 22 the loss coefficient at  $V_b/V_u = \infty$  increases as  $\theta$  decreases. The values at this limiting condition are represented by the correspondingly numbered dotted lines in Fig. 7. Since they are in the reverse order of the curves originating at the upper left, the curves must eventually cross. Consequently, the crossing of curves 2 and 3 is an explainable behavior.

It can also be deduced that the curves should have minimums. Consider a main pipe having a small aperture from which a stream issues at some angle  $\theta$ . If the flow is presumed one of constant mechanical energy the velocity diagram will be

that shown in Fig. 8. Here  $V_u$  is the component in the direction of the axis of the main. The velocity corresponding to the static pressure gradient,  $\sqrt{2g(P_u - P_b)}$ , is normal to  $V_u$ . Since the resultant is  $V_b$  the velocity ratio is simply

$$V_b/V_u = 1/\cos \theta \quad (23)$$

Therefore, there is a *natural* angle of discharge associated with each value of velocity ratio. The relationship is shown graphically in Fig. 9. When this natural angle coincides with the angle of the branch, an optimum condition, and hence a point of minimum loss, should be expected.

From Equation 23 or Fig. 9, anticipated minimum values for the curves of Fig. 7 are  $\infty$ , 2.00 and 1.41 for the angles of 90, 60 and 45 deg, respectively. From the curves themselves, curve 1 does not have a minimum in the range of the data whereas curves 2 and 3 have minimums at velocity ratios of 1.75 and 1.25, respectively. The agreement between the expected and actual minimums is, therefore, good.

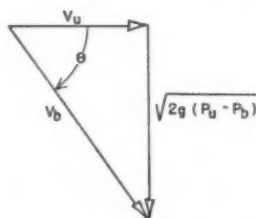


FIG. 8. VELOCITY DIAGRAM FOR FLOW FROM A SMALL APERTURE

That the actual minimums occur at somewhat lower velocity ratios than the calculated minimums can also be explained. The analysis leading to Equation 23 did not consider any change in the velocity in the main because the aperture was presumed to be very small. This condition certainly is not valid for the fittings of Fig. 7. It is, therefore, necessary to consider the effect of the size of the branch duct on the value of velocity ratio at which a minimum would be anticipated.

Referring to Fig. 2, consider the flow conditions along the entrance to the branch. At the upstream edge the velocity will be  $V_u$ , whereas the velocity at the downstream edge,  $V_d$ , will be less than  $V_u$  because the area of the main is constant and some of the flow has been diverted into the branch. Hence, along the entrance section the velocity will decrease in some manner from  $V_u$  to  $V_d$ . If the flow is one of constant mechanical energy, from Bernoulli's equation the pressure will rise correspondingly across the section. However, it is not necessary to restrict these considerations to an ideal fluid, because a velocity reduction and pressure rise will also occur with a real fluid, the difference being only the amount that the pressure will rise. With either fluid the net result is that at the downstream edge of the entrance section the velocity component of  $V_b$  along the axis of the main is less than that at the upstream edge; and the static pressure gradient, which yields the component normal to the axis of the main, is greater. As a consequence, the *natural*

angle of the stream at the downstream edge is greater than at the upstream edge. The minimum loss would, therefore, be expected to occur at some angle of discharge between these two extremes, and hence at an angle greater than calculated from Equation 23. Applied to a branch duct oriented at a specified angle, this means that this natural angle and the branch angle will coincide to form a condition of minimum loss at a value of velocity ratio *less* than that given by Equation 23. As previously discussed, the velocity ratios at the values of minimum loss for curves 2 and 3 of Fig. 7 are actually somewhat less than predicted; the experimental results are therefore in agreement with this concept.

Application of this concept to divided-flow elements having the same size of main duct and the same branching angle  $\theta$ , but different branch sizes, yields the conclusion that as the branch size is progressively increased the minimum values will occur at progressively lower velocity ratios. In air conditioning duct systems the branch size is ordinarily less than half the size of the main. Because of this and the fact that the slope of the loss curve will change slowly near the minimum, Equation 23 and Fig. 8 are considered of sufficient accuracy for engineering purposes.

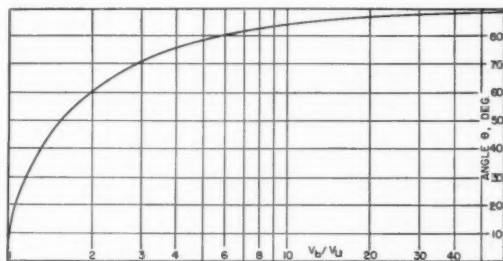


FIG. 9. NATURAL ANGLE OF EFFLUX

#### PREDICTED LOSS EQUATION

The most commonly used type of fitting is that with the branch perpendicular to the axis of the main; i.e.,  $\theta = 90$  deg in Fig. 2. For this type, from Equation 22 the loss of head at  $V_b/V_u = \infty$  is

$$H_b = 0.50 V_b^2/2g \quad (24)$$

An expression for the loss must satisfy this condition as well as the limiting condition at  $V_b/V_u = 0$  given by Equation 20. Moreover, since from Equation 23 the loss coefficient curve does not have a minimum at a finite velocity ratio, it must be either a continuously increasing or a continuously decreasing function. The simplest relation satisfying all three conditions is the sum of Equation 20 and Equation 24 because neither contributes to its opposite limiting condition. Thus,

$$H_b = 0.5 (V_b^2/2g) + (V_u^2/2g) \quad (25)$$

or, from the definition of  $\lambda_b$  given by Equation 16,

$$\lambda_b = 0.5 + (V_b/V_u)^{-2} \quad (26)$$

This relationship is shown as the dotted line, curve 4, in Fig. 10. The results of three independent laboratory investigations are also presented for comparison. Curve 1 has been obtained from the correspondingly numbered curve in Fig. 5 after converting from the  $\lambda'_b$  to the  $\lambda_b$  basis. Curve 2 shows the results of Vogel with a fitting geometrically similar to that of curve 1 but smaller (branch diameter of 0.59 in. vs. 3.49 in.). Curve 3 was obtained with a 12-in. x 8-in. main and a 7-in. diameter branch with air as the fluid. The agreement between curves 1 and 3, with water and air as the respective fluids, is excellent although the fittings were not geometrically similar. In addition, the agreement of these two curves with the pre-

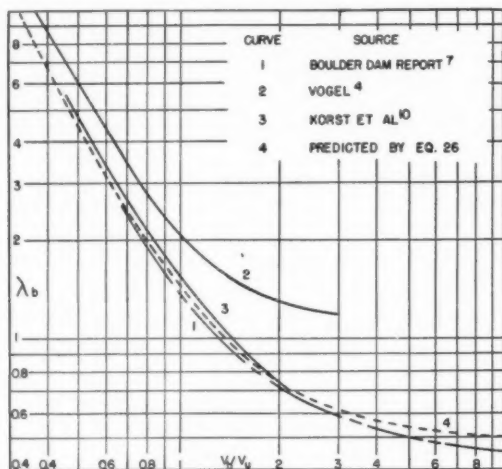


FIG. 10. RESULTS FOR 90 DEG BRANCHES AS OBTAINED BY DIFFERENT INVESTIGATORS

dicted equation given by Equation 26 is remarkable and evidently validates the method of analysis applied to the right-angled junction.

Deviation of curve 2 from the rest is apparently due to a scale effect; e.g., different relative roughnesses of the surfaces, resulting from the extremely small pipe sizes with which Vogel conducted his experiments. Since Korst<sup>10</sup> has shown that higher values of  $\lambda_b$  are to be expected at low Reynolds numbers, the fact that curve 2 is above the rest, rather than below, would be anticipated.

The predicted loss equation is also compared in Fig. 11 with the theoretical results of McNown<sup>9</sup>, from which it is seen that Equation 26 yields the same shape of curve but somewhat higher values. This is to be expected because the free-streamline theory presumes that the flow is one of constant mechanical energy (no loss in total head) up to the vena contracta of the jet in the branch. Taking this into consideration, the agreement between the curves is very good, especially for the more practical area ratio of 1/16.

It is hence concluded that Equation 26 accurately predicts the loss of head in the divided-flow fittings illustrated in Fig. 2 with a branching angle of 90 deg. For

branch ducts at other angles the condition of a minimum eliminates the possibility of determining a simple analytical expression to represent the loss curves. However, for engineering purposes values for the most frequently occurring angles can be obtained from Fig. 7.

It is also evident that the flow characteristics in the diverted-flow section of such divided-flow fittings can be explained rather clearly in a rational and conclusive manner.

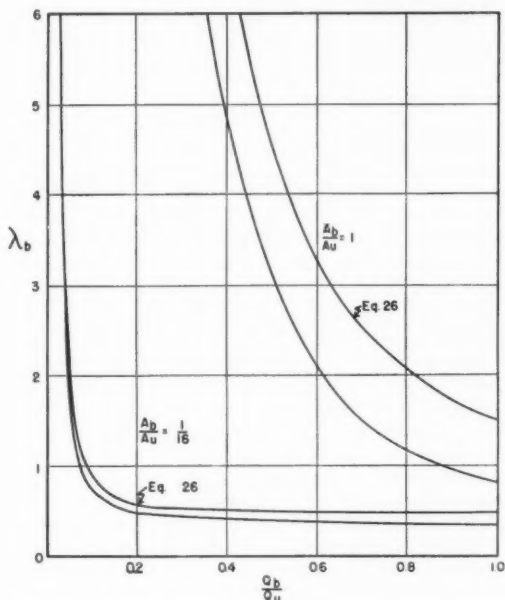


FIG. 11. COMPARISON OF PREDICTED AND FREE-STREAMLINE THEORY RESULTS

#### PRACTICAL INTERPRETATION OF RESULTS

From the loss coefficient curves for the type of fitting shown in Fig. 2, it is possible to develop practical information for designers of air duct systems, some of which will be presented. It is hoped that a thorough treatment of the practical aspects can be given in a future paper.

For air at the standard density of 0.075 lb per cu ft, Equation 11 becomes

$$H_d = 0.35 [(V_u - V_d)/4005]^2 \quad (27)$$

and Equation 25 becomes

$$H_b = 0.5 (V_b/4005)^2 + (V_u/4005)^2 \quad (28)$$

where  $H$  is now expressed in inches of water and represents a loss in total pressure, and  $V$  is in feet per minute.

Consider a main of constant cross section with several branch ducts at right angles. Since a portion of the total flow is diverted into the first branch the volume rate of flow in the main, and hence the velocity, will be reduced. As a consequence,  $V_u$  is progressively less at each branch. For the same velocity in each branch, the pressure loss in the diverted portion of the total flow (Equation 28) will also be successively less. Since the loss in the main (Equation 27) is small, if the distance between fittings is not great then the total-pressure loss of the first branch will exceed that of the combined main duct and last branch. This indicates that careful consideration should be given to the first few fittings during the design process. It also demonstrates that the commonly found instruction to *select the longest and most complicated run of duct and select the fan to overcome its pressure loss* can lead to an erroneous pressure requirement for the duct system.

That the losses in different branches may vary considerably is of particular importance to systems composed of several short branch ducts spaced relatively close together along a main. Suppose a main of constant area is to have several branches of equal lengths oriented at 90 deg to it. If the spacing between branches is small, the branch farthest downstream will have the highest flow rate while the one farthest upstream will have the lowest. On the other hand, if it were desired that each branch deliver the same cfm of air, the calculations would show that the upstream branch would have to be larger than the downstream branch.

As a further practical consideration one possible way of handling the same flow rate with the same size and length of branch is to vary the branching angle.

The fact that Fig. 10 shows the pressure loss to be more than 8 *branch velocity heads* at  $V_b/V_u = 0.35$  gives the impression that an extremely serious situation would exist if such low values of velocity ratio were experienced in a duct system. Such is not necessarily the case. For a given value of  $V_u$ , as  $V_b/V_u$  decreases  $V_b$  also decreases; but when the number of velocity heads *increases* rapidly the branch velocity head *decreases* even more rapidly and the net effect is the reverse of expectation. For example, with  $V_u = 2000$  and  $V_b = 2000$ ,  $V_b/V_u = 1.0$  and  $H_b = 0.375$ ; whereas for the same  $V_u$  and  $V_b/V_u = 0.35$ ,  $V_b = 700$  and  $H_b = 0.280$  in. of water. Thus it is seen that even though the loss is 8.65 velocity heads for the latter case the total-pressure loss in inches of water, instead of being an expected extremely large value, is actually 25 percent less than that for the expected low-loss condition.

Dimensionless plots typified by Figs. 10 and 11 are of value from a research standpoint but they are not essential to the design of duct systems and tend to give misleading impressions. Equations 27 and 28 give the respective losses in total-pressure for the main duct and the 90 deg branch of the Fig. 2 fitting, and these equations furnish all the information the application engineer needs.

## CONCLUSIONS

1. Analysis of divided-flow fittings is more complex than through-flow fittings because an additional variable is involved.
2. The pressure loss can arbitrarily be referred to either of two characteristic velocity heads; hence, care must be exercised when interpreting results of different investigations.
3. The behavior of the loss coefficient curves can be explained in a rational and conclusive manner.
4. Application of concepts developed yields a predicted equation for the pressure loss which is in excellent agreement with experimental results.

5. The results of the analyses provide practical information of direct aid to designers of air duct systems.

#### ACKNOWLEDGMENT

Much of the material in this paper has been taken from the author's doctoral thesis<sup>17</sup>. Acknowledgment is, therefore, made to Profs. H. H. Korst, W. N. Espy and S. Konzo of the Department of Mechanical Engineering of the University of Illinois, under whose guidance the thesis was written. Acknowledgment is also made to the Department of Mechanical Engineering of which Prof. N. A. Parker is Head, for permission to publish.

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## DISCUSSION

P. B. GORDON, New York, N. Y.: The author is to be complimented on this contribution to the total literature having to do with this field of endeavor, information that ultimately permits translation into application data.

You will note that it was pointed out that, as we tend toward higher velocities and as the velocity heads go higher, the need for conserving energy and the need for more accurate determination of fitting losses becomes of increasing and more serious importance. That which is not too important at velocities of a thousand to two thousand feet a minute becomes of extreme importance as we go to three and four thousand feet a minute and higher.

It also shows a need for some creative work that could be done in the field of designing or studying optimum fittings that would provide for the lowest possible pressure loss at the fitting.

JOHN EVERETTS, JR., Philadelphia, Pa.: This is a most interesting paper and, I believe, brings us to a point where we can properly evaluate pressure losses in duct outlets.

I would like to ask the author, in regard to Fig. 5, which shows the loss coefficients for the two divided-flow fittings, one a straight outlet and the other with a conical outlet, showing a ratio between the two types of about two to one, why all the other data and curves are predicated only on the straight outlet rather than the conical. It was mentioned that data from this paper are important in high velocity work, but no mention is made that it is far more important in a dual-duct system, which not only has high velocity but also varying quantities of air in the hot and cold duct, which accentuates losses as against those of a high velocity duct in which the  $X$  point is constant.

I think perhaps that conical fitting connection may be far more important in that type of system in relating our pressure losses than would be indicated in this paper.

I also wish to compliment the author on his last page here, in which he says "Practical interpretation of results". Very seldom do we get a technical paper before this Society in which we in the design field can apply this information to practical results, and that has been the criticism for a long time which has been thrown at the Committee on Research.

I would like to have that compliment sent back to the author.

AUTHOR'S CLOSURE: I agree wholeheartedly with Mr. Gordon's comment that the present trend toward higher duct velocities makes it imperative that 1) pressure losses of fittings be accurately known, and; 2) fittings having the lowest possible pressure losses should be developed specifically for high velocity systems.

In reply to Mr. Everetts, the reasons for emphasizing the straight, rather than conical, takeoff are twofold. First of all, the straight type is the one most commonly used and therefore of greatest immediate concern. Secondly, very little research has apparently been conducted on conical takeoffs, probably because of the additional fabrication cost involved and unawareness of its efficient performance. Mr. Everetts' compliment on interpreting the research results in terms of application engineering design data is sincerely appreciated.

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## PANEL COOLING FOR A RESIDENCE†

By R. R. IRWIN\*, STILLWATER, OKLA.

THE COOLING phase of the process of air conditioning always has been one of the major problems in widespread residential application. The equipment required to carry out the cooling is both complicated and expensive, and requires high-grade and expensive energy for operation. These factors have been largely responsible for the many attempts to provide cooling for human comfort without using refrigeration equipment.

A basic factor to be considered is that all refrigeration processes result in heat being transported from the space or material being cooled to some medium existing at a higher temperature level. The usual medium to which this heat is transferred is the earth or its surrounding atmosphere. This heat transfer is usually accomplished by the well known air-cooled, water-cooled and evaporative condensers. It is evident that the air-cooled and evaporative condensers discharge their heat directly to the medium, atmospheric air. Since most water-cooled condensers now use, or are of necessity being converted to, cooling towers as a source of cooling water, we might say that almost all refrigeration systems discharge their heat to this medium, atmospheric air. The size of equipment and amount of energy required to carry out this cooling process is a function of temperature of this medium, the atmospheric air.

Two different temperature levels are available in the atmospheric air. They are the dry-bulb and wet-bulb temperatures. The dry-bulb temperature is the air temperature as recorded on the standard thermometer and the maximum usually encountered in central Oklahoma ranges from 95 to 105 F. The wet-bulb temperature, which represents the depression of the temperature by evaporative cooling, ranges downward from around 78 deg in extremely hot weather in that same locality.

Many attempts have been made to furnish the cooling effect required in summer to provide comfort for the human body. It is obvious that atmospheric air at temperatures above the normal body temperature is not going to provide effective cooling. The lower wet-bulb temperature of the atmosphere has challenged many men to develop a simpler process than the refrigeration system that has long been used for that purpose.

The familiar evaporative cooler is an example of an attempt to apply the wet-bulb medium directly for cooling. These units have a major fault in that the nature of the process results in the addition of moisture to the space being cooled, while the

† This paper is the result of research by the School of Mechanical Engineering and the Division of Engineering Research, Oklahoma Agricultural and Mechanical College.

\* Associate Professor, Mechanical Engineering, Oklahoma Agricultural and Mechanical College.

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most desirable process is one that removes moisture. Evaporative systems were being used for many applications a few years ago, but, due to the limited comfort results obtained, they are being superseded in many areas.

### PANEL COOLING RESEARCH

Realizing the limitations of the usual evaporative cooling equipment, but reasoning that the wet-bulb temperature level was low enough to produce human comfort, the author initiated a research project in this field in 1949.

In this research, it was proposed to obtain cooling of a medium by circulating water, cooled by evaporation in a cooling tower, through a panel cooling system



FIG. 1. FLOOR PLAN OF THE APARTMENT

similar to the conventional hot water panel heating system. The panel cooling system was visualized as coils of pipe built into both the ceiling and walls of the test unit. By including the coils in both walls and ceiling a means would be provided to intercept the heat flowing into the space from the outside.

### PRELIMINARY HOT BOX TESTS

In the first stage of the investigation, a typical section of a frame building wall equipped with a pipe coil was tested in a *guarded hot box*. By testing the section in the *guarded hot box* any desired summer temperature could be simulated inside the box. Since the temperature of water from a cooling tower is a function of the outside air wet-bulb temperature, the tests were run with variable water temperatures in order to predict the overall performance of the cooling panel. These tests furnished values for inside wall surface temperatures to be obtained with various cooling water temperatures when the wall was subjected to a heat source similar to that caused by outside summer heat.

This preliminary work provided some useful guides for further investigation. The results indicated that the temperature of the water flowing through the panel coil was at all times lower than both the inside and outside wall surface temperatures, indicating that heat was flowing from both surfaces to the coil inside the

wall. The logical conclusion would be that a cooling panel of this type could stop successfully the flow of summer heat from outside to the interior of a building. It would also be logical to conclude that this same panel will have a cooling effect on the interior of a building.

#### THE TEST APARTMENT

The investigation with the *guarded hot box* left the impression that a building might have a heat gain from various sources in excess of the cooling ability of the system. However, the results were so encouraging that it was decided to expand the research, and conduct tests under actual living conditions in an apartment in



FIG. 2. OUTSIDE VIEW OF APARTMENT SHOWING ALSO THE COOLING TOWER

the college housing facilities. The unit selected (see Figs. 1 and 2) was a one-story frame structure about 20 ft square, isolated on one side by a covered breezeway about 10 ft wide and on another side by a storage room, with two weather-exposed walls. The floor consisted of a concrete slab on the ground. The stud spaces in the walls were filled with loose rock wool, and the ceiling was covered with the same material. The panel coils, consisting of  $\frac{3}{4}$  in. copper tubing on centers of 6 in., were mounted on the inside of the existing wall board. The coils were then covered with plaster to form new walls and ceiling for the interior. The installation of a water circulating pump and cooling tower completed the cooling system.

Since the purpose of the research was to determine the effect of the system on human comfort, it was decided that the apartment should be rented to tenants by the college with the understanding that the tenants would cooperate with the testing. Recording instruments were used to obtain data on the temperature and relative humidity, both inside and outside, and water temperatures on entering and leaving the panel coils. In addition, the occupants were interviewed at various times, and a number of people who visited the apartment were interviewed concerning their feeling of comfort.

#### TEST RESULTS

From the data recorded over a period of two years testing, Figs. 3, 4, 5, 6, and 7 have been prepared showing data of representative typical days from the standpoint

of summer weather conditions. Figs. 3 and 4 represent typical hot summer days with some variation of humidity. Fig. 5 represents the hot dry summer day, while Fig. 6 represents the cool dry day and Fig. 7 represents the cool humid summer day.

A feature noted as the test charts were studied was the small variation in the values of inside temperatures and humidities. Many of the test charts (not shown

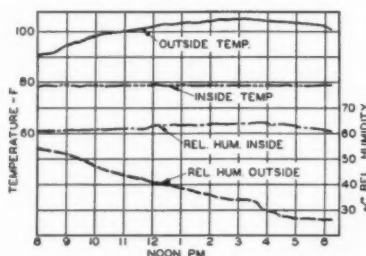


FIG. 3. INSIDE AND OUTSIDE AIR TEMPERATURES AND RELATIVE HUMIDITIES AS RECORDED ON A TYPICAL HOT SUMMER DAY

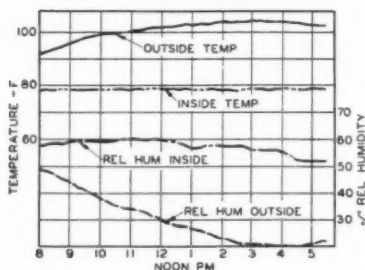


FIG. 4. INSIDE AND OUTSIDE AIR TEMPERATURES AND RELATIVE HUMIDITIES AS RECORDED ON A TYPICAL HOT SUMMER DAY

here) indicate inside temperature variations not to exceed 1 deg, while the outside temperature varied by as much as 20 deg.

The technical data obtained from the apartment test unit agree with the data of the preliminary *guarded hot box* investigation to a very great extent. However, the resulting inside atmospheric conditions from the apartment test and their effect on comfort were rather startling. The tests indicated that the inside temperature at no time during the whole period exceeded  $82\frac{1}{2}$  F with outside temperatures reaching as high as 106 F. Since no attempt was made to control the inside temperature of the apartment, the room temperatures varied from the maximum to the low

seventies, depending on the outside wet-bulb temperature. As was to be expected, the relative humidity in the room varied somewhat with the outside relative humidity, but not uniformly. The test instruments recorded a condition of around 70 percent RH (relative humidity) on muggy days. The occupants indicated that, while they felt cool, the air had a close, stagnant feeling. The use of an ordinary room fan provided enough air circulation to remedy this situation. The occupants' comfort reaction after two summers of testing ranged from *very comfortable* to *fairly comfortable*, but at no time was a feeling of discomfort reported. The reactions of the various visitors interviewed agreed in the main with the occupants' feelings.

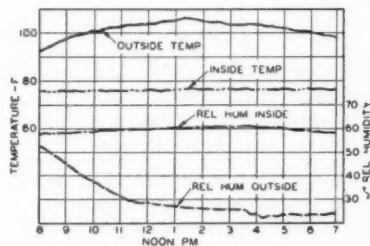


FIG. 5. INSIDE AND OUTSIDE AIR TEMPERATURES AND RELATIVE HUMIDITIES AS RECORDED ON A HOT DRY SUMMER DAY

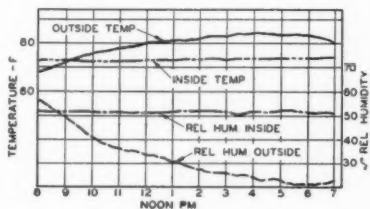


FIG. 6. INSIDE AND OUTSIDE AIR TEMPERATURES AND RELATIVE HUMIDITIES ON A COOL DRY DAY IN SUMMER

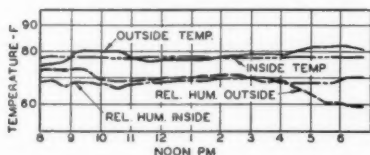


FIG. 7. INSIDE AND OUTSIDE AIR TEMPERATURES AND RELATIVE HUMIDITIES ON A COOL HUMID DAY IN SUMMER

## WHY NO DISCOMFORT?

Plotting values of inside temperatures and humidity on the *comfort chart* readily explains the comfort reactions of the people from *very comfortable* to *fairly comfortable*, but it does not explain the failure to register discomfort, as some of the values do not fall within the comfort zone. The explanation of the degree of comfort seems to lie in the effect of radiation on the heat loss of the human body. A widely used equation<sup>1</sup> gives some indication of the relation between air and surface temperatures and a feeling of comfort. The equation is

$$t_s + t_a = 160 \quad (1)$$

where

$t_s$  = mean surface temperature, Fahrenheit degrees.

$t_a$  = room air temperature, Fahrenheit degrees.

160 = a constant\* based on air and surface temperatures of 80 F.

A study of a typical residence cooled by circulating cool air will show that the temperatures of the inside surfaces of ceilings, exposed walls, windows and doors are higher than the room air temperature. Normally, the surfaces of the floors and partition walls are cooler than or equal to the room air temperature. If the mean value of these surface temperatures is greater than 80 F, then the room air must be maintained at a temperature below 80 F to give the desired feeling of comfort.

It would seem to follow then, that the factors of temperature and humidity of air and the temperatures of surrounding surfaces must be evaluated in determining the degree of comfort of a room. When this investigation was started, it was felt that the lack of dehumidifying equipment in the system would result in conditions of discomfort during muggy weather. The test data show that conditions of high humidity did occur, but the reactions of the occupants were not ones of discomfort, as expected from the comfort chart. In fact, the occupants at times expressed a preference for the conditions maintained in the apartment to those maintained in the usual air conditioned spaces. There can be no doubt that the radiant cooling effect of wall and ceiling panels offsets the warming effect of the high humidity.

This theory would seem to be borne out by other research<sup>2</sup> carried out at the University of Florida. A study of the use of water spray to cool roofs of buildings indicated that the feeling of comfort was improved by cooler ceiling temperatures with no accompanying reduction in the air temperature of the space.

## CONCLUSION

No attempt has been made in this report to present in detail the great mass of data collected or to describe the testing procedure in detail. It is recognized that all phases of the research have not been completed. Unfortunately, the testing facilities did not permit a full evaluation of the radiation effect on the occupants' comfort.

The use of cooling panels in rooms having a high internal heat gain and com-

<sup>1</sup> Radiant Energy Exchange as a Factor in Airplane Cabin Heating, by B. F. Raber and F. W. Hutchinson (*Journal of the Aeronautical Sciences*, July 1944).

\* For heating, this constant is usually stated as 140, but 160 is believed more nearly to indicate the relationship to the higher air and surface temperatures involved in cooling.

<sup>2</sup> Roof Spray for Reduction in Transmitted Solar Radiation, by G. E. Sutton (ASHAE TRANSACTIONS, Vol. 57, 1951, p. 321).

paratively small weather walls and ceilings would present a different problem from that encountered in the type of residence tested.

Air motion is an important factor in comfort control, and it seems unlikely that a panel cooling system without any means of circulating air would be as comfortable as desired. It is likely that some applications would have latent gains that would require dehumidifying equipment in addition to the panel cooling system.

However, the conclusion, based on the test data and the comfort reactions of the occupants, indicates that it is possible to provide summer comfort in a residential unit with a panel cooling system using a cooling tower as the only source of cooling effect, at least in the summers experienced in Oklahoma.

## DISCUSSION

JOHN R. WATT, Austin, Tex., (WRITTEN): Professor Irwin must be congratulated for performing this experiment. His results emphasize a principle long evident to many engineers in the Southwest:—air conditioning is possible in this region without the expensive equipment necessary in more humid areas.

Unfortunately, manufacturers design cooling equipment for the East, Southeast, and Middle West, not the Southwest. Thus, local engineers and contractors must usually forfeit the cooling possibilities afforded by the climate. Their customers thereafter consume 40-60 percent more power than necessary, taxing already overburdened utility companies.

What seems needed is further development of equipment like Professor Irwin's which uses evaporative cooling principles to cool rooms without adding humidity to the air enclosed. His failure to reach the "Comfort Zone" is accidental; the traditional ASHAE Comfort Chart ignores radiant cooling. The Bio-Climatic Chart developed by the Olgyay brothers,\* does not and would show Professor Irwin's true success.

Two experimental dry air evaporative coolers of 3-ton size are now being readied for testing at the University of Texas. Other experiments in the general field of *indirect* or *dry-air* evaporative cooling are detailed in a 1954 report from the University of Texas to the U. S. Naval Civil Engineering Research and Evaluation Laboratory, Port Huene, California. I believe copies are available upon application.

Professor Irwin's process is especially promising because it combines advantageously with radiant heating. It is hoped commercial development will arise from his pioneering.

ROBERT S. ASH, Phoenix, Ariz., (WRITTEN): This paper represents a valuable contribution to the use of the evaporative cooling process for comfort conditioning. The various methods of indirect evaporative cooling deserve much needed research.

In the Southwest, evaporative cooling of attic spaces to reduce the summer heat gain on air conditioning systems using mechanical refrigeration, have been tried with considerable success. On some installations, it has been possible to cool the structure with three tons of refrigeration where five tons would have been required if evaporative cooling of the attic space had not been practiced.

Professor Irwin mentions in his paper that evaporative systems are being superseded in many areas. I presume he is referring to the State of Oklahoma. On the contrary, more evaporative air coolers were installed in Oklahoma in 1954 than in any previous year. Actually Oklahoma is one of the largest users of evaporative coolers, exceeding in use states such as New Mexico and Arizona.

Although the prevailing summer wet-bulb temperature in Oklahoma is fairly high, direct evaporative cooling is popular. The coolers are usually sized to change the air every minute or every minute and one-half. This results in an air movement of about 100 fpm in the occupied space and is responsible for reducing the effective temperature

\* Application of Climatic Data to House Design, Division of Housing Research, Housing and Home Finance Agency, Washington, D. C.

an additional one to two degrees below that shown on a still air effective temperature chart.

In my opinion the results obtained with evaporative panel cooling versus the results that can be achieved with direct evaporative cooling, taking into account the economics involved, do not justify the panel cooling method. However, as a means of reducing the heat load on a mechanical refrigeration system, it has considerable promise.

H. E. DEGLER, Kansas City, Mo., (WRITTEN): This *guarded hot box* method to provide summer comfort in a residential unit with a panel cooling system and using a cooling tower as the only source of cooling effect is indeed a contribution to the literature.

Professor Irwin says that the inside temperature at no time exceeds 82½ F with outside temperatures as high as 106 F. For the conditions shown in Fig. 3, what was the temperature of the water to and from the cooling tower? How much was in the cooling system, and what was the rate of circulation?

It would be interesting to know what improvement could be shown in the cooling ability of this system if a mechanical draft cooling tower were used instead of a natural draft type. It is my opinion that this substitution would provide 3 to 5 degrees of additional cooling. I wish that Professor Irwin would comment on this substitution of type of cooling tower.

P. B. GORDON, New York, N. Y.: This is an interesting paper. It touches on four separate mechanisms in this overall concern of panel cooling and its relationship to the future of air-conditioning.

One, it touches on the panel cooling problem, itself, where the panels receive heat from the interior spaces after the heat has been released within the space, such as heat that enters through transmission, that which comes through by infiltration, or heat released within the space by lighting, other equipment, or people.

The second mechanism involves the deliberate creation of thermal storage without waiting for the building materials to drop. This second mechanism involves deliberately lowering the temperature of the building mass through low wet-bulb periods to prepare the mass to receive heat during higher wet-bulb periods.

The third mechanism is of extreme interest, that of the intercept or barrier that Professor Irwin pointed to.

The fourth concerns the overall one of surface temperature as related to comfort.

So we have these four interesting mechanisms and their interplay in this one paper.

I have a few comments that I would like to introduce. Number one, I would like to ask Professor Irwin if he has available a calculation for the heat gain to the space, assuming that it were to be handled by mechanical refrigeration.

Number two, I would like to ask if Professor Irwin has any idea as to what would happen if, instead of treating all the wall surfaces and the ceiling surface, he were to treat with panels the ceiling surface only? That is, what would happen as to the resultant inside temperature, and what would happen as to the intercept or barrier effect? Now, I am thinking of the next step forward: Instead of what is substantially a large one-room area, if one were to have a six or an eight room house, would it be necessary to provide for all interior surfaces or both sides of the interior surfaces in order to take advantage of the lower mean surface temperature, since there are now six surfaces to worry about? With only a ceiling surface available, it may not be possible to reduce the mean surface temperatures sufficiently to reach the same objectives.

JOHN EVERETTS, JR., Philadelphia, Pa.: This is an excellent piece of work that Professor Irwin has done, from one standpoint in particular. All of our work on comfort to date has been done without considering radiation. Its all been done on the basis of controlling the psychrometric conditions of air. We do know, from experience, that if we let the surfaces ride where they will, due to our heat gain thru walls and heat, pick-up from lights, that the limit of human tolerance is very small, only amounts to about one and a half or two degrees, from the optimum point of comfort.

When we got into panel cooling, we found, much to our surprise, that that limit of tolerance has extended to somewhere around four or five degrees. In other words, in one job we have quite an extensive panel cooling system; we can vary from 73 to 78 with less complaint of discomfort than in a straight air conditioning job, which we could only vary from 74 to 76.

This paper brings us to the other end of our problem. In other words, Professor Irwin is controlling the surface and is not giving any consideration to the control of the psychrometric conditions, which brings us to a limit which we feel is definitely of importance from the standpoint of comfort.

When Professor Irwin says that the test data show conditions of high humidity, but the reaction from the occupants was not one of discomfort as expected from the comfort chart, I'd like to point out that the comfort chart does not consider radiation effect. However, the program of the laboratory which will start sometime this fall, will take that into account to re-evaluate the comfort chart from the standpoint of radiation effect.

I think that will help us tie together these two differences of opinion, as to whether we should use all psychrometric controls on the one hand or all panel controls on the other hand, and to evaluate the differences that may come up between those two.

The only other comment I have is that from this photograph I should hope that if this type of system does become important for conditioning homes, the cooling tower may be made to look better.

G. V. PARMELEE, Cleveland, Ohio: Professor Irwin has presented a very interesting paper on a method of providing indoor comfort by use of a natural resource, the outdoor air, aided by relatively simple mechanical equipment. Because it is easy to install rather expensive air-conditioning equipment to provide indoor comfort, we have too often overlooked using our natural resources, as Professor Irwin has used them, to achieve our goals. I would like to see us take a greater interest in problems of this sort. I hope that Professor Irwin finds it possible to continue his investigations to further develop its potentialities.

Perhaps this system could profitably be adapted to commercial installations to remove some of the heat supplied by lighting installations. Panels could be supplied with sufficiently cool water to carry away a considerable amount of unwanted energy. There would be several practical problems to be considered, one of which is that of maintaining the water in good condition to prevent fouling of tubes. I would like to inquire of Professor Irwin if he encountered any trouble with algae, scale formation or any of the troubles associated with cooling-tower operation.

AUTHOR'S CLOSURE: I should have had a couple of people here taking down about five sets of notes to keep up with all these questions.

Obviously, we have to answer part of these questions in this way, to the effect that many of these are problems we realized would develop and we expect others to develop as further investigation proceeds. I might hazard opinions on some of these questions, but they are not based on any data we have collected.

As to whether or not we might omit some surfaces, any reply concerning this is purely an opinion, but I feel that we will need to provide all weather-surfaces with an intercepting barrier such as the ones in this project. I feel that to be true because if one considers the levels of temperatures, water and air, and compares them with the probable panel temperatures, it is observed that there is a very small temperature differential. This means that as large a surface as possible must be available to absorb the heat.

I don't recall the exact cooling load calculation but believe it was in the order of about two tons of cooling for this particular project.

I intended to indicate earlier the rate of water circulation, which was measured at 28 gpm. The range of water temperature was from as little as two degrees to as much as six or seven degrees. In the early spring and late fall, the temperatures of both inside-air and water drop considerably. In fact, the occupant let the system operate into rather cool weather in the fall, and not knowing just how to shut the system off, had to start the

heating system up to offset the cooling effects of our panel cooling system. So that is a factor to be considered.

The cooling tower was a natural draft tower, made available through the courtesy of a tower manufacturer. It was rather large for the requirement, because we had asked to be furnished with a tower that would give about a two degree spread between wet-bulb and leaving-water temperatures. This being pretty close, the manufacturer was generous in the size of the tower.

I don't know that we got quite that small a temperature difference because the water circulation may not have been up to the quantity required by the tower to give that small a temperature difference, but it was a quite small temperature range and it's possible that mechanical draft cooling towers with a closer approach to wet-bulb would reduce those temperatures, but not too much below results obtained, because we didn't have the usual seven to ten degrees split found in commercial cooling tower applications.

In regard to water treatment, we did not treat the water, and the water didn't accumulate any slum and dirt, aside from the atmospheric dust blowing in, and this was removed by cleaning out the tower occasionally. It is possible there will be algae and scale formation. Very likely we were just fortunate in avoiding it.

The tower used in the system is still being used, although we haven't gathered any additional test data in the last two summers; but it's still being used by occupants of that particular apartment. In fact, it is somewhat of a choice unit in the college housing. When the tenant gets ready to move, other people are around talking to him, trying to lease the apartment.

We used a three-quarter horsepower motor on the pump and its current in-put was something under that maximum, about the equivalent of a half horsepower.

However, one of the primary factors that we were concerned with in this system, from the commercial consideration, is the operating cost. One feature of this system is that a fairly large amount of cooling effect is provided at a comparatively low operating cost.

The system may also be used as a conventional heating system, in fact this particular apartment unit has the heating equipment installed in conjunction with it, but that was simply done in order to make the apartment usable the entire year, and to prevent any freezing up of the piping during the winter time. We weren't particularly trying to do any investigative work in panel heating, because we felt that was pretty well covered by other people.

Referring to comments of Mr. Everetts, we too found that the limits of air temperature affecting comfort were extended due, we feel, to the radiation effect. One reason no attempt was made to control the temperature of the system was that we could not determine which temperature should be subjected to control.

The comments of Mr. Gordon express very clearly the factors that affected the degree of comfort obtained in our research. All four of those factors plus air motion must be considered in the evaluation of comfort. The question raised in regard to the number of surrounding surfaces needing to be cooled, as well as some of the other questions, will have to await further and expanded research.

It is apparent from the comments of Mr. Ash and Professor Watt that many are meeting the challenge of need for research in this field.

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## WEATHER DATA ANALYSIS FOR COOLING SYSTEM DESIGN

Illustrated by Data for New Orleans

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with Tulane University, New Orleans, La.

THE NEED for improved summer weather data for use in cooling-load estimating has prompted several analyses of weather records since the work of Albright<sup>1</sup> was published in 1939. His design values were based upon separate or single-element frequency analyses of dry-bulb, wet-bulb and dew-point temperatures. Sunshine and wind data were also included. In 1941 Kroeker and Soballe<sup>2</sup> demonstrated a method of making a two-element frequency analysis using dry-bulb and dew-point temperatures. In 1948<sup>3</sup> new design values of dry-bulb temperatures were published in THE GUIDE. These were based upon a frequency analysis made by the U. S. Weather Bureau covering 5 years of record for June, July, August, and September in over 100 U. S. cities. In 1947 a detailed analysis<sup>4</sup> of 5 years of record for Detroit was completed by the ASHAE Technical Advisory Committee on Weather Design Conditions. A two-element frequency analysis of summer dry-bulb and wet-bulb temperatures plotted on a psychrometric chart was included.

Development of the concept of sol-air temperature, which combines solar radiation, dry-bulb temperature and wind velocity in a single meteorological element, led to analyses,<sup>5,6</sup> published in 1945, of summer weather data and sol-air temperatures for New York City and Lincoln, Nebr. Further application of the sol-air temperature concept was suggested and the TAC on Cooling Load<sup>7</sup> recommended that data for New Orleans be studied. An analysis of these data is the subject of this paper.

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<sup>1</sup> Exponent numerals refer to References.

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## SOURCES OF DATA

Dry-bulb and dew-point temperatures, wind speed and wind direction used in this analysis are from records of hourly observations made by the U. S. Weather Bureau at the New Orleans airport station. The station is located 11 miles northwest of the city at 29 deg 56 min north latitude and 90 deg 7 min west longitude. These data are for the months of June, July, and August and cover the 10-year period from 1932 through 1941, the same period of record used for analysis of the New York and Lincoln data. The solar radiation data were observations made by Drs. H. S. Laurens and H. S. Mayerson at the Tulane University campus, as hourly and daily totals of direct and diffuse radiation incident upon a horizontal plane. Measurements were made with an Eppley pyrheliometer, located on the roof of a laboratory building approximately 100 ft above sea level in an atmosphere relatively free of smoke and dust.

## SELECTION OF SUMMER DESIGN CONDITIONS

Ordinarily, it is not economical to provide sufficient cooling capacity to maintain indoor design conditions under the worst possible weather conditions that might be encountered in a given locality. Usually somewhat less severe weather conditions are selected as a design basis, so that indoor dry-bulb and humidity will exceed their design values on a few days during the cooling season. The frequency

TABLE 1—COOLING LOAD ANALYSIS—THREE DIFFERENT BUILDINGS IN NEW ORLEANS\*

LOAD	BTU PER HR	PERCENT OF TOTAL LOAD	PERCENT DUE TO WEATHER
<b>RESIDENCE:</b>			
Transmission.....	29000	38.3	38.3
Solar.....	10200	13.4	13.4
Lights.....	3500	4.6	....
People.....	1600	2.1	....
Ventilation.....	31500	41.6	41.6
<b>TOTALS.....</b>	<b>75800</b>	<b>100.0</b>	<b>93.3</b>
<b>OFFICE BUILDING:</b>			
Transmission.....	285000	3.8	3.8
Solar.....	340000	4.4	4.4
Lights.....	1408000	18.5	....
People.....	1687000	22.3	....
Ventilation.....	3840000	51.0	51.0
<b>TOTALS.....</b>	<b>7560000</b>	<b>100.0</b>	<b>59.2</b>
<b>THEATER:</b>			
Transmission.....	63000	2.7	2.7
Solar.....	132000	5.7	5.7
Lights.....	35000	1.5	....
People.....	840000	36.1	....
Ventilation.....	1260000	54.0	54.0
<b>TOTALS.....</b>	<b>2330000</b>	<b>100.0</b>	<b>62.4</b>

\* From Reference 7.

with which this can be tolerated influences the selection of design weather conditions.

Selection of design weather conditions is also influenced by the type and use of the structure to be air conditioned. The proportion of the total system load that can be attributed only to the weather varies from one installation to another. Moreover, different weather elements have varying degrees of importance in the weather's effect on the load.

This point is illustrated by cooling load analyses of typical buildings in New Orleans made by Everetts<sup>7</sup> (See Table 1). In the example of the residence in that table, the cooling load attributable to the weather constitutes about 93 percent of the total load. The dry-bulb temperature and sun (transmission and solar) account for 51.7 percent while the wet-bulb temperature and wind (ventilation or infiltration) account for 41.6 percent. On the other hand, of the total load on a theater cooling system, the weather accounts for about 62 percent, of which 54 percent (ventilation) is attributable to the wet-bulb temperature.

Finally, the selection of design conditions depends upon the thermal capacity of the building. Large internal thermal capacity introduces time lags and damping effects in the conversion into cooling system load of instantaneous rates of heat gain due to sun and DB temperature.

It, therefore, appears that design weather conditions are best selected by the design engineer after he has made a critical study of each case. With this in mind a multi-element frequency analysis was made from which an engineer can select an appropriate *design day* for his case.

#### FREQUENCY ANALYSIS

*Method of Analysis:* Weather for a one-day period was defined by the maximum dry-bulb and the mean dew-point temperatures for the day and the total solar radiation incident upon a horizontal plane during the same day. Dew-point temperatures were used because wet-bulb temperatures were lacking. The latter would have been preferable.

Details of the analysis are as follows. For each day when the maximum dry-bulb temperature was 90 F or greater, the maximum dry-bulb temperature, the mean dew-point temperature and the total solar radiation incident upon a horizontal plane were tabulated. Arbitrary ranges for each of the three values were then chosen and the tabulated data were classified. A 3-deg range in dry-bulb and mean dew-point temperatures and a 400 Btu per (sq ft) (day) range in solar radiation were used. Obviously, other ranges could be used. (In a recent analysis of summer dry-bulb temperatures Court<sup>8</sup> used a 5-deg range.)

*Results of Analysis:* Analysis of the New Orleans data by this method led to the three-element frequency distribution given in Table 2. Various statistics of interest can be determined from this table. For example, there were 32 days in all, or 3 per season, when the maximum dry-bulb temperature exceeded 95 F, regardless of mean dew-point and total solar radiation. These days occurred more frequently with middle values of solar radiation than with extremely high or extremely low values, and most of them occurred with mean dew-points in the 73-75 F range. Other analyses can be made from Table 2 by the engineer and proper weight can be given to air enthalpy (a function of wet-bulb temperature), solar radiation and dry-bulb temperature to suit each design.

Note that there are 421 days, or 45.6 percent of the period, represented in Table 2 with maximum dry-bulb temperatures of 90 F or greater. There are an addi-

TABLE 2—THREE-ELEMENT FREQUENCY ANALYSIS OF SUMMER WEATHER DATA FOR NEW ORLEANS

(For months of June, July and August, 1932 through 1941)

TOT. SOLAR RADIATION ON HORIZONTAL PLANE [BTU/(SQ FT) (DAY)]	MEAN DEW-POINT F DEG	NUMBER OF OCCURRENCES, DAYS			TOTAL DAYS	PERCENT OF PERIOD
		MAX DRY-BULB, F DEG				
		90-92	93-95	>95*		
More than 2800	61-63	1			1	0.11
	64-66	1			1	0.11
	67-69	1		2	3	0.33
	70-72		3		3	0.33
	73-75	1		2	3	0.33
	76-78				..	..
Total days		4	3	4	11	
Percent of period		0.44	0.33	0.44		1.21
2400-2800	61-63				..	..
	64-66	2			2	0.22
	67-69	3	1	1	5	0.54
	70-72	3	1	1	5	0.54
	73-75	7	16	2	25	2.72
	76-78	3	2		5	0.54
Total days		18	20	4	42	
Percent of period		1.95	2.17	0.44		4.56
2000-2400	61-63	1			1	0.11
	64-66	3	3		6	0.65
	67-69	10	6	1	17	1.85
	70-72	11	6	1	18	1.95
	73-75	19	25	7	51	5.55
	76-78	1	3		4	0.44
Total days		45	43	9	97	
Percent of period		4.90	4.67	0.98		10.55
1600-2000	61-63		1		1	0.11
	64-66		1		2	0.22
	67-69	5	6		11	1.17
	70-72	10	6		16	1.74
	73-75	42	24	6	72	7.83
	76-78	8	13	1	22	2.39
Total days		65	45	7	117	
Percent of period		7.06	4.90	0.76		12.72
1200-1600	61-63				..	..
	64-66				..	..
	67-69	4			4	0.44
	70-72	8	6	1	15	1.63
	73-75	44	13	3	60	6.52
	76-78	13	11	3	27	2.93
Total days		69	30	7	106	
Percent of period		7.50	3.26	0.76		11.52
Less than 1200	61-63				..	..
	64-66				..	..
	67-69				..	..
	70-72	5			5	0.54
	73-75	28	2	1	31	3.37
	76-78	6	6		12	1.31
Total days		39	8	1	48	
Percent of period		4.24	0.87	0.11		5.22
Totals for all values of Total Solar Radiation	61-63	2	1		3	0.33
	64-66	6	3		9	0.98
	67-69	23	8	4	35	3.80
	70-72	37	22	3	62	6.74
	73-75	141	80	21	242	26.38
	76-78	31	35	4	70	7.61
Total days		240	149	32	421	
Total Percent		26.09	16.20	3.49		45.78

\* Figures in this column are for temperatures above 95.

tional 5.3 percent of days not included because of lack of solar data but for which the maximum dry-bulb temperature was 90 F or greater. In this period the maximum observed dry-bulb temperature was 101 F, the maximum dew-point was 85 F and the maximum wet-bulb was 86 F.

A fourth weather element, wind velocity, should be included as a design condition. Study of the records showed that the 421 days included in Table 2 were about equally divided between days when the average wind velocity was between 4 and 6 mph and days when the average was between 6 and 8 mph. These differences are small, so that it was concluded that, for New Orleans, a fourth element need not be included in the frequency analysis. This might not be true for other localities.

From a load estimating viewpoint, wind velocity operates in 2 ways. An increase in wind velocity increases infiltration and so increases loads. Opposed is the fact that an increase in velocity increases the heat carried *away* from a surface which has absorbed solar radiation.

#### EXAMPLE OF SELECTING DESIGN CONDITIONS

A three-element frequency analysis of the type given by Table 2 provides the design engineer with sufficient information to make his own selection of design conditions for a particular building. Use of Table 2 will be demonstrated here by two *illustrative examples*, Case I, a Residence, and Case II, a Theater.

*Case I—Residence:* The cooling-load estimate for a residence would seem to require the selection of a combination with fairly high values of all weather elements, with emphasis on dry-bulb temperature and solar radiation. Consider the combination of 93–95 F dry-bulb and 73–75 F mean dew-point temperatures coincident with total solar radiation in the range of 2000 to 2400 Btu per (sq ft) (day).

Examination of Table 2 shows that there were 32 days with maximum dry-bulb temperatures above the 93–95 F range, 53 with solar radiation above the 2000–2400 Btu per (sq ft) (day) range and 70 days with mean dew-points above the 73–75 F range. However, the combination was exceeded in severity only 32 times or 3 days per season. For example, on 7 of the 32 days the mean dew-point was in the 73–75 F range and solar radiation in the 2000–2400 range, but maximum dry-bulbs were greater than the 93–95 F range. In all, the combination was equalled or exceeded in severity 57 times or an average of 6 days per season. Hence, a selection of design conditions which combines 93–95 F dry-bulb temperature, 73–75 F mean dew-point temperature and 2000–2400 Btu per sq ft total daily solar radiation, might be judged suitable for this case. These same conditions should also be suitable for load calculations for rooms on the sunny sides of an office building.

*Case II—Theater:* Table 1 shows that the weather factor of the cooling-load estimate for a theater is mainly the wet-bulb temperature (excluding internal load). Sun effects are of minor importance. Hence, the main factors to be considered are dry-bulb and dew-point temperatures. Table 2 shows that dew-points in the range 76–78 F occurred 70 times in the 10-year period or 7 days per season on the average when the maximum dry-bulb was 90 F or greater. Days with dew-points in the 76–78 F range coincident with a maximum dry-bulb temperature in excess of 95 F occurred only 4 days in the 10-year period, but there were 35 days when the dry-bulb temperature was in the 93–95 F range, and 31 when maximum dry-bulbs were in the 90–92 F range. Conservatism would probably dictate

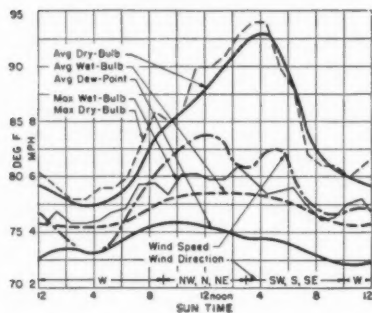


FIG. 1. DRY-BULB, DEW-POINT AND WET-BULB TEMPERATURES, WIND SPEED AND WIND DIRECTION AT NEW ORLEANS FOR CASE I WEATHER CONDITIONS

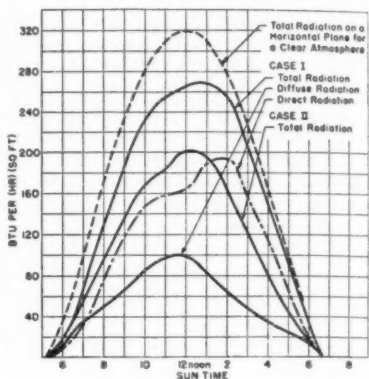


FIG. 2. SOLAR RADIATION ON A HORIZONTAL PLANE AT NEW ORLEANS FOR CASE I AND CASE II WEATHER CONDITIONS

selection of the 93–95 F range to be used with the 76–78 F dew-point range. High solar intensities rarely occurred with high dew-point and a selection of a range as low as 1200–1600 might be judged adequate for use in load calculations.

#### DIURNAL CYCLES

It should be evident that to be of maximum use, design conditions should go beyond a statement of how often a certain value, or a combination of single values of several elements, is likely to occur. This is all that Table 2 shows. Design conditions should also include *diurnal cycles* that are associated with the selected design values. Hence, an attempt to develop these cycles for the two illustrative examples was made.

**Case I—Temperatures:** This case has dry-bulb temperatures in the 93–95 F range, mean dew-point in the 73–75 F range, and total solar radiation in the 2000–2400 Btu per (sq ft) (day) range. Cycles were prepared by averaging hourly values of the days in this category. The results are shown as curves in Fig. 1 and as values rounded off to the nearest whole number in Table 3.

The dry-bulb curve is similar to the curves of individual days. As might be expected, the averaging process yields a lower daily range for the average curve, 15 deg, as compared to 16.5, the average of the individual daily ranges. The maximum in the set of days selected was 19, the lowest 14 deg.

The daily range of the dew-point curve, 4 Fahrenheit degrees, is 2 degrees lower than the average daily range. The maximum range was 10, the minimum 4. The maximum dew-point in this combination was 78 F, the minimum was 68 F.

The wet-bulb temperature curve of Fig. 1 was determined by averaging hourly values of computed wet-bulb temperatures. The maximum hourly value for the days averaged was 80.5 F, the lowest was 71.3 F. The curves made up of straight-line segments give the temperatures for the day with highest wet-bulb temperatures of the 14 days averaged.

TABLE 3—SUMMER DRY-BULB, DEW-POINT AND WET-BULB TEMPERATURES AT  
NEW ORLEANS, FAHRENHEIT DEGREES  
(For the Period 1932 through 1941)

SUN TIME	AVERAGE CONDITIONS						MAXIMUM OBSERVED
	CASE I			CASE II			
	DRY BULB	DEW POINT	WET BULB	DRY BULB	DEW POINT	WET BULB	DRY BULB
1 a.m.	79	73	76	79	75	76	87
2	78	74	76	79	75	77	85
3	78	74	75	79	76	77	85
4	78	73	75	80	76	77	85
5	78	74	76	80	75	77	84
6	79	74	76	81	75	77	85
7	80	75	76	82	76	77	90
8	83	76	77	84	77	79	92
9	84	76	78	86	78	80	94
10	85	76	78	87	79	80	94
11	87	76	78	89	79	81	96
12	88	76	79	90	79	81	97
1 p.m.	89	75	79	92	77	80	97
2	91	75	79	93	76	80	98
3	92	75	79	92	76	80	98
4	93	75	78	91	76	80	99
5	92	74	78	88	76	79	101
6	90	74	78	85	76	77	99
7	86	74	77	83	75	77	96
8	83	73	77	81	75	77	92
9	81	73	76	80	75	76	88
10	80	72	76	79	75	76	89
11	80	72	76	79	75	76	89
12	79	72	76	79	75	76	88
Avg.	83.8	74.2	77.0	84.1	76.2	78.1	..
Daily Range	15	4	4	14	4	5	..

*Case I—Wind Speed and Direction:* Hourly wind speed and direction for Case I are shown in Fig. 1. The 24-hr average speed of 5.5 mph is only a little less than 10-year averages of 6.2, 5.8 and 5.6 mph for June, July, and August, respectively. The maximum daily average for the Case I combination was 6.0 mph and the minimum was 3.5 mph. It is of interest to note the coincidence of change in wind speed with change in wind direction. Data such as these would be particularly significant in the application of natural ventilation, though local differences may be more important than these characteristics of a single location.

*Case I—Solar Radiation:* The solar radiation records were in the form of *totals of solar radiation* received on a horizontal plane in one hour's time. Each total was listed under the hour for the end of the period. Hence, each value represented approximately the *intensity of radiation for the mid-point of the one-hour period*. Totals for each hour of the Case I days were averaged and plotted. A smooth curve resulted, from which intensity values in Btu per (hour) (square foot) were

taken. These values are plotted in Fig. 2 and listed in Table 4. Also listed in Table 4 are maximum hourly intensities in June, July and August.

Computed values of the direct and diffuse components for a horizontal surface are shown in Fig. 2. Values of vertical-surface total incident solar radiation have been computed with these data as a starting point and are given in Table 5. Fig. 2 also gives the *clear atmosphere* values for a horizontal surface computed from Table 4, Chapter 13 of THE GUIDE 1954. Note that the radiation received per day on a square-foot horizontal surface is 2558 Btu for a clear atmosphere, or 19 percent more than for Case I conditions.

TABLE 4—SUMMER INTENSITY OF TOTAL SOLAR RADIATION INCIDENT UPON A HORIZONTAL SURFACE AT NEW ORLEANS  
(For the Period 1932 through 1941)

SUN TIME	AVERAGE CONDITIONS		MAXIMUM OBSERVED		
	CASE I	CASE II	JUNE	JULY	AUGUST
	Intensity in Btu per (hr) (sq ft)				
6 a.m.	18	18	76	101	52
7	67	54	154	165	110
8	127	92	239	220	192
9	184	135	282	283	258
10	231	166	344	341	305
11	255	184	375	376	338
12	264	202	385	382	355
1 p.m.	269	196	393	369	348
2	255	164	384	366	320
3	207	123	340	325	296
4	151	80	286	241	218
5	91	47	229	191	144
6	35	15	134	122	96
Approx. Total, Btu per (sq ft) (day)	2154	1476	3621	3482	3032

In order to compute the intensity of radiation on vertical surfaces, it was necessary to know the direct beam component of the total solar radiation which falls on the horizontal surface. Research<sup>9</sup> at the ASHAE Research Laboratory has developed a method by which these components can be determined for cloudless skies. Comparison of the Case I total of 2154 Btu per (sq ft) (day) was made with an analysis of cloudless day totals by Fritz.<sup>10</sup> He showed that June, July and August totals for New Orleans on such days were 2510, 2440 and 2240 Btu, (sq ft) (day). Therefore, it was concluded that the total of 2154 represents, on the average, a day of little cloudiness. Consequently, the ASHAE cloudless day data on diffuse radiation were used in determining solar radiation components for New Orleans. Sun positions corresponding to August 1, 30 deg north latitude and 90 deg west longitude were used in these computations.

*Case II—Diurnal Cycles:* Fig. 3 illustrates average dry-bulb, dew-point, and wet-bulb temperatures for Case II, which represents days with maximum dry-bulb temperatures in the 93–95 F range, mean dew-points in the 76–78 F range and total solar radiation in the 1200–1600 Btu per (sq ft) (day) range. Values rounded

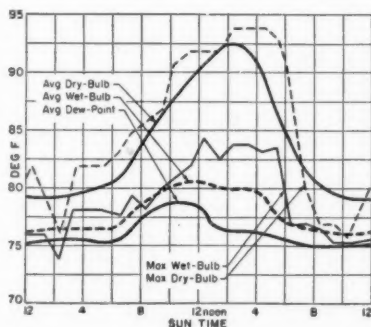
TABLE 5—TOTAL SOLAR RADIATION INTENSITIES (DIFFUSE PLUS DIRECT) ON HORIZONTAL AND VERTICAL SURFACES FOR CASE I COMBINATION OF WEATHER CONDITIONS COMPARED  
(For August 1, 30 deg north latitude)

SUN TIME	SURFACE ORIENTATION				
	HOR.	N	E	S	W
	BTU PER (HR) (SQ FT)				
6 a.m.	18	21	58	7	5
7	67	23	123	15	11
8	127	29	169	25	17
9	184	29	165	43	22
10	231	31	134	62	28
11	255	33	85	74	32
12	264	35	37	80	37
1 p.m.	269	32	31	77	90
2	255	28	27	64	149
3	207	25	22	43	191
4	151	28	17	21	211
5	91	41	10	12	197
6	35	36	5	5	122

off to the nearest whole number are listed in Table 3. The dry-bulb curve of this figure would be similar to that of Fig. 1 if it were shifted two hours to the right. The dew-point and wet-bulb curves of Fig. 3 could be roughly approximated by adding 2 degrees to the corresponding curves of Fig. 1. Whether or not the differences in corresponding curves are significant can be learned only by examining more data. Wet-bulb and dry-bulb temperatures for that day in this combination, which had the highest wet-bulb values, are shown for comparison. The main differences in wet-bulb temperatures occur between 11:00 a.m. and 6:00 p.m.

Values of total solar radiation for Case II are shown in Fig. 2 and listed in Table 4. Total radiation received of 1476 Btu per (sq ft) (day) is about 69 percent of the Case I total. The 24-hour average wind speed was found to be 5.9 mph, not significantly greater than for Case I. The maximum daily average was 7.3 mph, the minimum was 4.5 mph.

FIG. 3. AVERAGE AND MAXIMUM DRY-BULB AND WET-BULB TEMPERATURES AND AVERAGE DEW-POINT TEMPERATURES AT NEW ORLEANS FOR CASE II WEATHER CONDITIONS



## SOL-AIR TEMPERATURES

Heat transfer into the weather side of a building surface is a result of a complex energy exchange between the building surface and the outdoor environment. Heat-transfer calculations are further complicated by periodic variations of the various weather elements, but they can be greatly simplified by application of the concept of the sol-air temperature.<sup>11</sup> This temperature can be used to compute the heat flow into a building surface and is defined by Equation 1 (see Reference 12):

$$t_e = \frac{\alpha I}{h_o + h_r} + \frac{\epsilon(R - \sigma T_o^4)}{h_o + h_r} + t_o \quad \dots \quad (1)$$

where

$$h_r = \frac{\epsilon \sigma (T_o^4 - T_e^4)}{t_o - t_e}$$

and where

- $\alpha$  = absorptivity of the building surface for solar radiation, dimensionless.
- $\epsilon$  = emissivity of the building surface for low-temperature radiation, dimensionless.
- $\sigma$  = Stefan-Boltzmann constant =  $0.173 \times 10^{-8}$  Btu per (hr) (sq ft) (F deg abs<sup>4</sup>).
- $h_o$  = convective conductance at outdoor surface, Btu per (hour) (square foot) (Fahrenheit degree).
- $I$  = incident solar radiation, Btu per (hour) (square foot).
- $R$  = incident low-temperature radiation from the outdoor surroundings, Btu per (hour) (square foot).
- $t_o$  = sol-air temperature, Fahrenheit degrees.
- $t_o, T_o$  = dry-bulb temperature of outdoor air, Fahrenheit degrees and Fahrenheit degrees absolute, respectively.
- $t_e, T_e$  = temperature of building surface, Fahrenheit degrees and Fahrenheit degrees absolute, respectively.

Equation 1 shows that the sol-air temperature consists of three components as follows:

1. An equivalent temperature for the short-wave solar radiation absorbed by the surface.
2. An equivalent temperature for the long-wave low-temperature radiation exchange with the outdoor surroundings.
3. The dry-bulb temperature of the air.

*Sol-Air Temperatures for Case I:* Components of sol-air temperatures for Case I weather conditions are listed in Table 6 and are based upon average values of 1.10 Btu per (hr) (sq ft) (F deg) for  $h_r$  and 2.50 for  $h_o$ . The low temperature radiation component was evaluated by adaptations of Equations 4 and 6 of Reference 12.

For any hour the sol-air temperature is the algebraic sum of three components, except that the solar radiation component must be multiplied by the value of  $\alpha$  appropriate to the surface color and condition.

For example, at 2:00 p.m., the sol-air temperature for a flat roof with  $\alpha = 0.9$  is  $0.9 \times 71 - 6 + 91 = 148.9$  F; at 2:00 a.m.,  $t_e = -5 + 78 = 73$  F.

## SUGGESTIONS FOR FURTHER STUDY

The diurnal cycles developed for two different combinations of weather elements raise questions as to their similarity, or lack of it, to curves for other combinations;

TABLE 6—COMPONENTS OF SOL-AIR TEMPERATURES FOR HORIZONTAL AND VERTICAL SURFACES FOR CASE I COMBINATION OF WEATHER CONDITIONS, FAHRENHEIT DEGREES

SUN TIME	DRY BULB COMPONENT	HORIZONTAL SURFACE		VERTICAL SURFACES				
		SOLAR RADIATION COMPONENT <sup>a</sup>	LOW TEMP RADIATION COMPONENT	SOLAR RADIATION COMPONENT <sup>a</sup>				LOW TEMP RADIATION COMPONENT
				NORTH	EAST	SOUTH	WEST	
1 a.m.	79		-5					-2
2	78		-5					-2
3	78		-5					-2
4	78		-5					-2
5	78		-5					-2
6	79	5	-5	6	16	2	1	-1
7	80	19	-6	6	34	4	3	0
8	83	35	-6	8	47	7	5	0
9	84	51	-6	8	46	12	6	0
10	85	64	-6	9	37	17	8	0
11	87	71	-6	9	24	21	9	0
12	88	73	-6	10	10	22	10	0
1 p.m.	89	75	-6	9	9	21	25	0
2	91	71	-6	8	8	18	41	0
3	92	58	-6	7	6	12	53	0
4	93	42	-6	8	5	6	59	0
5	92	25	-6	11	3	3	55	0
6	90	10	-6	10	1	1	34	0
7	86		-6					-1
8	83		-6					-2
9	81		-6					-2
10	80		-6					-2
11	80		-6					-2
12	79		-5					-2
24-hr avg.	83.8	24.9	-5.7	4.4	10.2	6.1	12.9	-1.0

<sup>a</sup> This component is evaluated for  $\alpha = 1.0$  and must be multiplied by the value of  $\alpha$  appropriate to the surface color and condition.

first, for New Orleans, and second, for other cities or regions. Further study might lead to simple methods of expressing diurnal cycles from daily indices of the various weather elements.

Desirable improvements would be the use of observed wet-bulb temperatures instead of mean dew-point temperatures and the use of a simple daily index of this element.

Finally, attention needs to be given to the possibility of design conditions chosen from a frequency analysis such as Table 2 being exceeded in severity for a large number of days in one season. Such an example occurred in June 1936, when there were 25 days on which the maximum dry-bulb temperature was 90 F or greater and the total solar radiation was in excess of 2400 Btu per (sq ft) (day). On 11 of these days, the total was in excess of 2800, accounting for all of the values in the highest bracket of solar radiation in Table 2.

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## DISCUSSION

J. P. STEWART, Syracuse, N. Y., (WRITTEN): This paper is an excellent presentation of data and a method to select design conditions for the major weather elements affecting cooling load estimates.

For New Orleans the solar radiation for Case I, for a residence or office building, was explained to be 19 percent less than the design data for clear skies published in the ASHAE GUIDE, compared on the basis of daily total solar radiation. Is the major portion of this difference due to the presence of clouds? Some care must be exercised in applying reduction factors for clouds. Table 5 shows that there were considerably more clouds or haze in the morning than in the afternoon. For Case I design conditions, the solar radiation through an East window at 8:00 a.m. would be 28 percent less than the ASHAE GUIDE design value whereas the solar radiation through a West window at 4:00 p.m. would be only 11 percent less than the GUIDE value. Is this condition characteristic only of New Orleans, or do the authors think that there are generally more clouds in the morning for all areas in general?

In Table 1, the office building appears to have very little glass area because the percentage solar load is about the same as the theatre. Usually office buildings have fairly large window openings and consequently large solar loads.

Full advantage should be taken of the reduced solar radiation due to clouds in deter-

mining the total refrigeration load on a building consisting of several different exposures, but the writer wonders whether it is safe to take full advantage of the clouds when selecting equipment for handling the individual zones such as the sensible heat load only on the exterior zone of office buildings where there is a very large solar load.

It is hoped that other similar analysis can be made in the Mid-west plains area, such as Dallas, Texas, and in the Great Lakes region or on the Eastern seacoast.

JOHN EVERETTS, JR., Philadelphia, Pa.: As Chairman of the TAC on Weather Data, I would like to say that there seems to be quite a bit of confusion between what that committee is supposed to accomplish and what the other TAC's are supposed to get from this weather data, from the standpoint of this paper.

In other words, now I'll have to wear two hats, one as a design engineer and the other as the Chairman of the TAC on Weather Data.

As a design engineer, the only comment I have is the one which George also brought out, and that is that I would rather wet-bulb be used instead of dew-point because we have far more data on wet-bulb, or will have, than on dew-point.

The question as to this particular paper is that its a tool by which the design engineer can make a selection of weather data for a particular job. There is no quarrel about that. As a matter of fact, the Weather Data Committee has been instructed by the other TACs that what is wanted in the way of weather data is a breakdown of how long, in terms of percentage, various temperatures exist in various cities. A year ago a questionnaire was sent out which some took a lot of time to fill out. This was certainly appreciated. There were 245 returns out of 600 questionnaires—a terrifically remarkable return.

On the basis of those returns, next January at the Annual Meeting, it is expected that the complete analysis of winter dry-bulb data will be presented, broken down from 1 percent probability of the time up to 50 percent probability. The Heating Load Committee can then tell designers how to use those data. Its not up to the Weather Data Committee to determine what design temperature to use or how to use it. That is up to the Heating Load Committee on heating and to the Cooling Load Committee on cooling. This paper is an expression thru the Cooling Load Committee as to how weather data can be applied. We are striving as rapidly as we can to get weather data, (not design data, but weather data) to you, so that these other committees can determine how to apply the data in design. I might say also that we are trying to overcome the objection of this 5-year period, the 10-year period, the 3-year period, the 1-year period, that some people have been talking about. We are planning to present the data on the basis of whatever number of years the records are available in a given weather station. If the Oklahoma City weather station has been in operation for 27 years there will be 27 years of data. To get 37 years of data it will be necessary to wait another ten years and then the Committee will propose it but now all that can be done is to present what is available. Data for some cities go up to almost 90 years, and will be available; so there will be no question of 5-, 10-, or 15-year period of cycles, and whether it's hotter or colder this sun mer than last, or what have you. It's going to be analyzed data from some eight million tabulator cards the Weather Bureau has on file, and which they are grinding out as rapidly as they can for us.

The only objection I have to this particular paper is the use of dew-point data instead of wet-bulb data, which I think should be corrected. Otherwise, I think this is a very important tool to use for design where one wants to select or be selective in his design for temperatures and solar radiation.

**AUTHORS' CLOSURE (Mr. Parmelee):** I would like to emphasize that this paper's chief feature is a method of analyzing weather data and putting it in such form that a design engineer can make his own selection of design weather conditions. To illustrate the use of the data assembled, illustrative selections of design conditions have been made for two different types of buildings. One should not, however, draw the conclusion that these selections are to be taken as design conditions for New Orleans.

Mr. Stewart asks if clouds are responsible for the fact that the solar radiation intensities of the Case I (residence) selection are lower than the design data for clear skies as

given in the ASHAE GUIDE. Comparison of the total radiation received per day with data of Fritz for cloudless days in New Orleans shows that there could not have been much cloudiness, so that on the whole, haziness is probably responsible. Records of some of the days averaged in Case I showed, however, that clouds must have obscured the sun at times. Mr. Stewart points out that the Case I selection shows that the morning values of solar intensity are lower than the afternoon values, because of greater cloudiness or haziness in these hours, and asks if this is true for all areas in general. It is probably not true for all areas, though it was true for New Orleans for this level of solar intensity and the associated dry-bulb and dew-point temperatures. Because dry-bulb and dew-point temperatures reflect atmospheric conditions, the level of these values may be tied in with the shape of the solar radiation curves. More data would have to be analyzed. Note that Table 5 shows that for the Case II selection of design weather conditions, the morning solar intensities are higher than the afternoon values, just opposite to the Case I values. In this case there was considerable cloudiness. Hence, it is quite unsafe to generalize.

Mr. Stewart points out that full advantage should be taken of the reduction in solar intensities by clouds when one is calculating the load for an entire building, if it has several different exposures. He doubts that this should be done in figuring equipment for individual zones which have large solar loads. We believe that this is for the individual engineer to decide, and believe that data in the form of Table 2 provides some, at least, of the necessary information on which to base a judgment. In general, it seems advisable to design equipment for individual zones to handle clear-day solar loads but to base the total load on somewhat lower solar values. The type of system and zoning to be used must be considered, however. Perhaps some general rules could be developed if weather data were available in sufficient detail that a number of different cases could be analyzed.

Mr. Everetts' comments on this paper and on weather-data analysis in general are much appreciated. We agree that wet-bulb temperature should be used instead of dew-point in weather analysis for heating and cooling, especially since it is the basic observation. Moreover, it is not subject to "rounding-off" errors as is dew point. It should be noted that the local monthly climatological data supplements now issued by the U. S. Weather Bureau give wet-bulb temperatures observed on each day at 1 a.m., 7 a.m., 1 p.m., and 7 p.m. Presumably values for the other hours are readily available for analysis.

The question of period of record was discussed by Mr. Everetts. For purposes of illustrating a method of analyzing and presenting weather data for design engineers, the 10-year period used in this paper was sufficient. This period was selected by the TAC on Cooling Load to coincide with the two analyses made by Professor Mackey some years ago. In developing design data, a longer period would be highly desirable and would be essential if attention were to be given to determining the number and length of periods of severe weather.

In conclusion, we wish to emphasize Mr. Everetts' observation that the paper presents a tool for the design engineer to use. Like any new tool, it should be tested; and, if found wanting, it should be improved or replaced by a better tool.



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## AN ANALYSIS METHOD FOR PREDICTING BEHAVIOR OF SOLID ADSORBENTS IN SOLID SORPTION DEHUMIDIFIERS†

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with The Pennsylvania State University, University Park, Pa.

THE OBJECTIVE of this paper is to present a method for predicting behavior characteristics of dynamic adsorption systems. Specifically, it aims at enabling the dehumidification machine designer, manufacturer or user to estimate the effect of changes in operating conditions on adsorption and desorption characteristics. The method is empirical in that test data from the system in question must be used in the formulation of the prediction equations, and the method is approximate in that certain mathematical assumptions are made to preclude the necessity of conducting an extremely large number of tests to obtain a complete picture of the variation of the adsorption characteristics (such as breakpoint useful concentration) with the primary variables (such as temperature and relative humidity of the flowing air).

It is realized that these assumptions impose limitations on the prediction technique. It is felt, however, that the savings in labor and time, as well as the increase in utilitarian value which results from these assumptions, serve to justify them, especially in this particular field where no behavior prediction method has yet been evolved, and where, because of the very large number of variables involved and the difficulty in getting experimental reproducibility, an approximate solution is all that presently can be hoped for.

Before proceeding with the analysis method, the so-called adsorption and desorption characteristics will be described and defined as they relate to dynamic dehumidification.

### CHARACTERISTICS AND DEFINITIONS

The only requisites for a dynamic dehumidifier are a desiccant bed, a fan to force the humid air through this bed and a heater to periodically reactivate the

† This research work has been carried on under the guidance of the ASHAE Technical Advisory Committee on Sorption. Personnel: G. L. Simpson, chairman; John Everetts, Jr., vice chairman; Gunnar C. F. Asker, Warren E. Emley, Jr., Oliver D. Colvin, Albert S. Gates, Jr., F. C. Dehler, E. W. Gifford, E. R. McLaughlin.

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adsorbent. As the air passes into the activated desiccant, it surrenders a certain amount of its water vapor. The rate of moisture pickup and the humidity condition of the leaving air are functions of a great many variables, some of which will be discussed later. The ratio of the amount of water adsorbed by the desiccant in a given time to the amount of water vapor in the air entering the desiccant bed during that time is known as *adsorption efficiency*. A characteristic of adsorbents in dynamic use is that this adsorption efficiency remains constant and at a relatively high level from the beginning of an adsorption cycle until some later point in the cycle at which time the efficiency begins to drop. This point is known as the *breakpoint*, and the time from the beginning of adsorption to this point is known as the *breakpoint time*. In the ideal case the breakpoint would coincide timewise with the sudden upswing of the effluent dew-point temperature. Although additional drying can be effected beyond the breakpoint, good commercial practice dictates that the gel be regenerated in the near vicinity of this point. Adsorption carried beyond the breakpoint continues at an increasingly slower rate until the adsorbent is completely saturated. This point is known as *completion*.

Adsorbents, such as silica gel and activated alumina, contain, even when reactivated, a small amount of water. This is usually about 5 to 7 percent of the dry weight of the desiccant and is known as residual moisture. Attempts to remove it result in a physical change in the substance and a reduced adsorptive capacity. The term *useful concentration* is used to designate the percent of moisture in the adsorbent over and above the residual quantity. This percentage is generally based on the dry weight of the desiccant.

When regeneration of the adsorbent is desired, the heater is energized and the direction of air flow through the bed is usually reversed. The effluent dry-bulb temperature rises rapidly at first, and then virtually levels off for a period of time. This period of level or slowly increasing temperature represents the period during which the major portion of the heat input is being used to boil off the adsorbed water. This temperature pattern continues until most of the water contained in or on the desiccant is released. Then, when the latent heat requirements begin to diminish, the heat input goes into sensible heat gain to the passing air stream. This is reflected in a rather sharp increase in the effluent dry-bulb temperature. This point, measured from the beginning of desorption, has been designated *temperature-rise time*. Although additional regeneration (at a slower rate) can be attained by continuing the heat addition process beyond the temperature-rise time, once again good commercial practice calls for reactivation to be ended in the near vicinity of this point. Regeneration past this point until the adsorbent is in moisture equilibrium with the air stream is known as *complete desorption* or *desorption to completion*. The energy expended in the heater per unit weight of water desorbed for any given time is called *economy of desorption* and usually has the dimensions of kw-hr/lb of water desorbed.

#### OBJECTIVES OF ANALYSIS METHOD

These two processes, dynamic adsorption and dynamic desorption, which have just been briefly outlined, are extremely difficult to analyze or predict, because they are transient and because they are functions of a great number of variables. Notable analytical work has been done both theoretically and empirically on adsorption and related phenomena, and almost all of this has appeared in chemical literature.

Although most engineering operations involving desiccants are of the dynamic and near-adiabatic variety, there is no accepted analysis method, to the knowledge of the authors, for predicting the behavior characteristics of desiccants under these conditions. In approaching the problem of evolving such a method for consideration, it was necessary to define the objectives desired and to evaluate properly the following observations:

1. All purely theoretical analyses to date have been handicapped by a multitude of simplifying assumptions, by assumptions as to the mechanism of adsorption, by not accounting for the possibility of more than one such mechanism, by insurmountable, or at best, extremely tedious, mathematical obstacles.

2. For engineering application a theoretical analysis is not requisite, nor, in some cases, even desirable.

3. The empirical approaches have necessarily been restrictive in range of operating conditions, as well as in applicability to specific adsorbents and adsorbates.

4. In no known case does either a theoretical or an empirical method of analysis enable the engineer to predict many of the design factors he most wants, such as adsorption efficiency, breakpoint time and cost of operation.

5. Numerous past experiments carried out in chemical laboratories by competent investigators using precision equipment have, unfortunately, made evident the fact that the variables involved are so numerous and the adsorbents themselves so inconsistent that close experimental reproducibility is seldom achievable. Because of the type and size of the equipment and the operating conditions, engineering tests are likely to be even less accurate. Also, laboratory-type experimental results are not necessarily valid for engineering application.

In the light of the aforementioned considerations a new analysis method should be, first, utilitarian both in relative ease of use and in telling the engineer what he wants to know (even if at a sacrifice of mathematical or physical rigor), and secondly, it should be based on engineering-type experimental results and applicable in principle to any adsorbent-adsorbate combination for which suitable data are available.

#### PREMISES OF PROPOSED METHOD

The proposed prediction method is based on the following premises:

1. That the adsorption characteristics desired are (a) completion useful concentration, (b) breakpoint useful concentration, (c) complete adsorption time, (d) breakpoint time and (e) adsorption efficiency to breakpoint.

2. That these five adsorption characteristics are, for a given system, primarily a function of the dry-bulb temperature and relative humidity of the air entering the bed, the rate of air flow and the bed-depth to bed-diameter ratio.

3. That the desorption characteristics desired are (a) completion useful concentration desorbed, (b) complete desorption time, (c) temperature-rise time and (d) economy of desorption to temperature-rise.

4. That these four desorption characteristics are, for a given system, primarily a function of the dry-bulb temperature and relative humidity of the air entering the bed, the rate of air flow per unit of heater output, the bed-depth to bed-diameter ratio, and the heater wattage.

Then, in general notation,

$$X = f(Y_1, Y_2, Y_3 \dots Y_n) \dots \dots \dots (A)$$

from which it is assumed that the functions are such that

$$dX = (\partial X / \partial Y_1) dY_1 + (\partial X / \partial Y_2) dY_2 + (\partial X / \partial Y_n) dY_n \dots \quad (B)$$

where

$X$  = desired adsorption or desorption characteristic.

$Y_1, Y_2$ , etc. = factors on which  $X$  is primarily dependent.

Now, by following a test procedure in which standard values are chosen for each primary variable and in which all variables except one are held at their standard values for any given test, it is possible to experimentally determine functions for the parenthetical partial differential terms in Equation B. In other words, if  $X$  is plotted against  $Y_1$ , the data being obtained from tests in which  $Y_2, Y_3, \dots Y_n$  were held at their standard values, a curve results which can be put into equation form and which, in turn, can be differentiated, with respect to  $Y_1$ , to yield  $(\partial X / \partial Y_1)$ .

Using this technique for each of the primary variables involved, an expression for  $dX$  is obtained. Assuming this to be integrable, the resulting expression for  $X$  will contain a constant of integration for each of the primary variables.

Actually, this so-called integration constant can, for a given variable, be either a pure constant, zero-valued, or some function of the other variables involved. The analysis method being proposed uses the assumption that the sum of the integration constants for each differential equation can, over a limited range of primary variable variation, be approximated by a pure constant, this constant being determined from the experimental data. This assumption has two basic and opposite ramifications; first, it renders the technique practical in that it enables the derivation of empirical prediction equations based on data from relatively few experimental tests; secondly, it imposes the most severe limitation on the efficacy of the results by not fully accounting for the intervariability of the primary variables. Because of the effects of this assumption, the *standard value* of each variable should be chosen at or near its normal or typical design level, and this should be in the vicinity of the middle of its test range. Similarly, it should be realized that reliability of prediction will generally decrease as the primary variables used in the prediction equations digress from their standard values.

#### METHOD APPLIED TO TEST DATA

The analysis method, having been explained in general notation, will now be further illustrated by applying it to actual test data. Although the method itself is universally applicable, it must be emphasized that the prediction equations that are evolved are for a particular adsorption system and must necessarily be based on a certain minimum amount of data from that system. In other words, the ultimate results achieved from data for a water vapor-activated alumina system will be different from the results reached through data from a water vapor-silica gel system, even though the same general technique is used in arriving at these results. There also would be differences in final prediction equations for widely different adsorption (or dehumidification) machines even though the same adsorbent-adsorbate combinations were used in each.

The data used in the analysis application to follow were obtained using commercial grade, 6-16 mesh silica gel as adsorbent and air-borne water vapor as adsorbate. The adsorption machine (desiccant tester, in this case) is shown in

Fig. 1. It was designed to be as simple structurally and functionally as possible so that there would be a minimum of outside influence on the behavior of the adsorbent under test. The desiccant bed was round with a cross-sectional area of 2 sq ft, and air flow was axial.

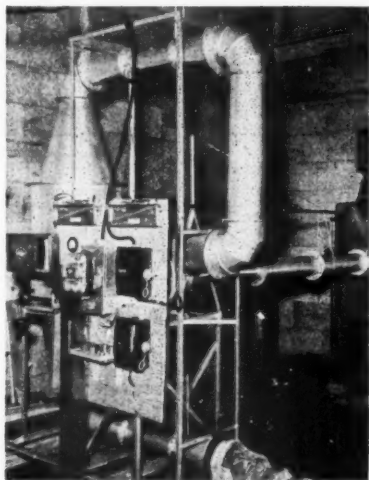


FIG. 1. TEST EQUIPMENT

A total of 24 tests was conducted, 11 adsorption and 13 desorption. In every case all primary variables except one were held at their respective *standard* values

#### NOMENCLATURE

- $B$  = depth-to-diameter ratio of adsorbent bed.
- $C$  = air flow, cubic feet per minute.
- $C_W$  = air flow to heater wattage ratio, cubic feet per minute per watt
- $E_B$  = adsorption efficiency to breakpoint, percent.
- $G$  = dry weight of adsorbent, pounds.
- $K$  = constant.
- $L$  = depth of adsorbent bed, inches.
- $M_R$  = economy of desorption to temperature-rise, kilowatt hours per pound of water desorbed.
- $R$  = relative humidity of air entering bed, percent.
- $T$  = dry-bulb temperature of air entering bed, Fahrenheit degrees.
- $t_B$  = breakpoint time (adsorption), minutes.
- $t_D$  = completion time (desorption), minutes.
- $t_R$  = temperature-rise time (desorption), minutes.
- $U_B$  = useful concentration at breakpoint, percent.
- $U_C$  = useful concentration at completion (adsorption), percent.
- $U_D$  = useful concentration desorbed, percent.
- $V$  = face velocity of air stream, feet per minute.
- $W$  = heater wattage.

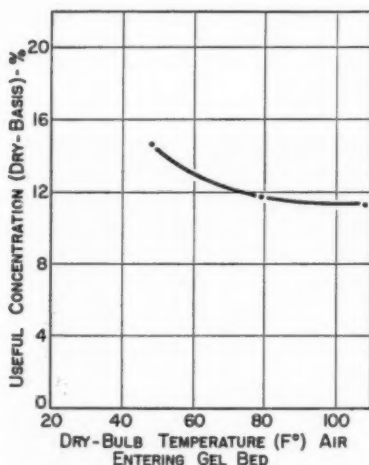


FIG. 2. EFFECT OF INLET TEMPERATURE ON USEFUL CONCENTRATION

for a given test. Conditions were near-adiabatic in that no after-cooling was employed. All tests were continued to completion. The air temperature entering the bed averaged 8-10 deg higher than room temperature, owing to a sensible heat gain through the fan.

TABLE 1—NOMINAL VALUES OF TEST VARIABLES  
(Face velocity in fpm equals cfm/2)

TEST VARIABLES	NOMINAL VALUES
Room Dry-Bulb Temperature, F. ....	40, 70 <sup>a</sup> , 100
Room Relative Humidity, percent .....	20, 35 <sup>a</sup> , 55, 90
Bed Thickness, inches .....	2, 4 <sup>a</sup> , 8
Adsorption Air Flow, cfm .....	60, 110 <sup>a</sup> , 170, 240
Reactivation Air Flow, cfm .....	50, 62.5, 67.5, 90, 112.5 <sup>a</sup> , 125
Heater Power, watts .....	2000, 2500, 4500 <sup>a</sup> , 5000
Cfm per Watt .....	0.015, 0.020, 0.025 <sup>a</sup>

<sup>a</sup> Chosen standard values.

A detailed description of the desiccant tester, test schedule (Table 1), procedure and experimental data will not be recounted here. This information is on file at the Main Library, The Pennsylvania State University<sup>1</sup> and at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS.

The method of obtaining an approximate prediction equation which has just

<sup>1</sup> Dynamic Characteristics of a Solid Chemical Desiccant, by W. L. Ross (Doctoral Thesis, Department of Mechanical Engineering, The Pennsylvania State University, June 1954).

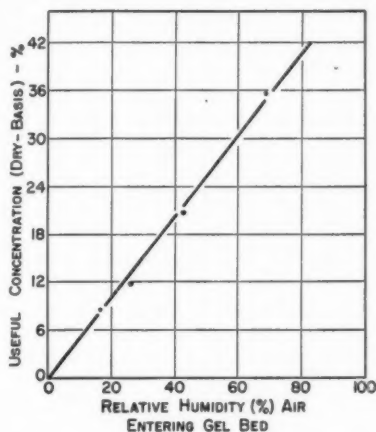


FIG. 3. EFFECT OF INLET RELATIVE HUMIDITY ON USEFUL CONCENTRATION

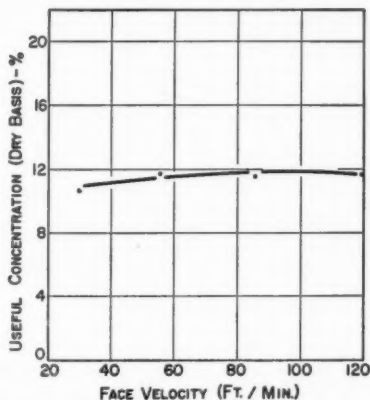


FIG. 4. EFFECT OF FACE VELOCITY ON USEFUL CONCENTRATION

been described in general terms will now be illustrated using completion useful concentration (adsorbed) as the desired characteristic of the dynamic system. Figs. 2-5 were plotted from the test results, and Equations 1-4 are analytical approximations to the curves shown in these figures. See Nomenclature for definition of the symbols used.

From Fig. 2,

$$U_o = -0.714 T + 0.0076 T^2 - 0.00002744 T^3 + 34.24 \quad (1)$$

From Fig. 3,

$$U_o = 0.505 R \quad (2)$$

From Fig. 4,

$$U_o = 0.00842 V + 10.9 \quad (3)$$

From Fig. 5,

$$U_o = 0.1375 L + 11.4 \quad (4)$$

Proceeding now with the design equation derivation,

$$U_o = f(T, R, C, B)$$

$$dU_o = (\partial U_o / \partial T) dT + (\partial U_o / \partial R) dR + (\partial U_o / \partial C) dC + (\partial U_o / \partial B) dB \quad (5)$$

From Equation 1,

$$\partial U_o / \partial T = -0.714 + 0.0152 T - 0.00008232 T^2 \quad (6)$$

From Equation 2,

$$\partial U_o / \partial R = 0.505 \quad (7)$$

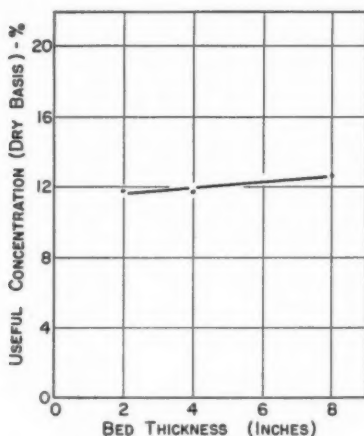


FIG. 5. EFFECT OF BED THICKNESS ON USEFUL CONCENTRATION

Since the face area of the bed was 2 sq ft, then  $C = 2V$  and Equation 3 can be written as

$$U_e = 0.00421 C + 10.9 \quad (8)$$

and from Equation 8,

$$\delta U_e / \delta C = 0.00421 \quad (9)$$

Since the bed diameter was 19.18 in., then  $L = 19.18 B$ , and Equation 4, can be written as

$$U_e = 2.6373 B + 11.4 \quad (10)$$

and from Equation 10,

$$\delta U_e / \delta B = 2.6373 \quad (11)$$

Substituting Equations 6, 7, 9, and 11 into Equation 5 yields

$$dU_e = (-0.714 + 0.0152 T - 0.00008232 T^2) dT + 0.505 dR + 0.00421 dC + 2.6373 dB \quad (12)$$

Assuming that the functions involved are continuous and that each has an indefinite integral,

$$U_e = -0.714 T + 0.0076 T^2 - 0.00002744 T^3 + K_1 + 0.505 R + K_2 + 0.00421 C + K_3 + 2.6373 B + K_4 \quad (13)$$

Combining the four constants of integration in Equation 13 and taking their sum to be approximated by a pure constant,

$$U_e = K - 0.714 T + 0.0076 T^2 - 0.00002744 T^3 + 0.505 R + 0.00421 C + 2.6373 B \quad (14)$$

Next, apply this equation to all of the adsorption tests conducted to determine the value of  $K$ . For example, in Test No. 1 the following conditions prevailed:

$$\begin{aligned}U_a &= 11.77 \text{ percent} \\T &= 79.0 \text{ F} \\R &= 25.8 \text{ percent} \\C &= 110.6 \text{ cfm} \\B &= 0.20855\end{aligned}$$

Substituting these values into Equation 14 yields

$$K = 20.23$$

Results of the substitution from all of the tests are given in Table 2.

TABLE 2—DETERMINATION OF EMPIRICAL CONSTANT,  $K$

TEST NO.	EXPERIMENTAL $U_a$		REQUIRED $K$
1	11.77	$= K - 8.48$	20.23
3	11.59	$= K - 7.55$	19.14
5	11.77	$= K - 6.90$	18.67
7	10.70	$= K - 8.94$	19.64
9	8.55	$= K - 13.21$	21.76
11	20.50	$= K + 0.01$	20.49
13	35.47	$= K + 13.28$	22.19
15	14.62	$= K - 4.48$	19.10
17	11.40	$= K - 8.14$	19.54
21	11.84	$= K - 8.52$	20.36
23	12.66	$= K - 7.59$	20.25

The average of the required  $K$ 's is 20.12, so the final prediction equation becomes

$$U_a = 20.12 - 0.714 T + 0.0076 T^2 - 0.00002744 T^3 + 0.505 R + 0.00421 C + 2.6373 B \quad (15)$$

Using this same technique with the appropriate curves from the test data, the following eight additional design equations were evolved:

*Breakpoint Useful Concentration*

$$U_B = -4.60 + 0.3323 T - 0.00321 T^2 + 0.00000895 T^3 - 0.0391 R + 0.00875 C - 0.000281 C^2 + 0.000001056 C^3 - 2.3975 B \quad (16)$$

*Completion Time per Pound of Adsorbent*

$$t_G/G = 83.7 - 2.914 T + 0.03143 T^2 - 0.0001148 T^3 + 0.2799 R + 127.63(C)^{-0.625} \quad (17)$$

*Breakpoint Time per Pound of Adsorbent*

$$t_B/G = 4.83 - 0.01836 T + 2.781 (0.707)^{R/10} - 0.0588 C + 0.0002968 C^2 - 0.0000004952 C^3 \quad (18)$$

*Adsorption Efficiency (to Breakpoint)*

$$E_B = 168.3 - 3.625 T + 0.0555 T^2 - 0.0002923 T^3 - 0.279 R + 24.4 (0.716)^{C/20} + 159.96 B - 577.191 B^2 + 487.696 B^3 \quad (19)$$

*Completion Useful Concentration Desorbed*

$$U_D = 8.08 - 0.275 T + 0.0017 T^2 - 0.0000029 T^3 + 0.494 R + 40.182 B - 127.836 B^2 + 132.296 B^3 \quad (20)$$

*Complete Desorption Time per Pound of Adsorbent*

$$t_D/G = 10.34 - 0.01478 T + 0.2031 R - 0.004884 R^2 + 0.00004157 R^3 \\ + 1.274 (B)^{-0.4618} - 8.98 (W/1000) + 2.326 (W/1000)^2 \\ - 0.205 (W/1000)^3 \dots \dots \dots (21)$$

*Temperature-Rise Time per Pound of Adsorbent*

$$t_R/G = 6.546 - 0.00573 T + 0.02383 R - 0.0001683 R^2 - 25.312 C_W \\ - 3.6162 (W/1000) + 0.83244 (W/1000)^2 - 0.064767 (W/1000)^3 \dots (22)$$

*Economy of Desorption to Temperature Rise*

$$M_R = 5.15 - 0.0628 T + 0.0006063 T^2 - 0.000001875 T^3 - 0.0231 R \\ + 0.0001778 R^2 - 19.46 C_W - 7.8964 B + 21.6125 B^2 \\ - 19.1071 B^3 - 0.14332 (W/1000) - 0.025 (W/1000)^2 \\ + 0.008333 (W/1000)^3 \dots \dots \dots (23)$$

## SOLUTION OF SAMPLE PROBLEMS

Several sample problems will now be solved through the use of the foregoing prediction equations.

*Example 1:* For a certain dehumidification job the air to be dried will enter the desiccant (6-16 mesh silica gel) at the design conditions of 80 F dry-bulb and 45 percent RH (relative humidity). The dehumidifier has a cylindrical bed 18 in. in diameter and 6 in. deep.

Estimate the maximum rate of air flow that can be used if adsorption efficiency (to breakpoint) must be maintained at 85 percent or higher.

*Solution:* Refer to Equation 19. Substitution  $T = 80$ ,  $R = 45$ ,  $B = 6/18 = 0.333$  and  $E_B = 85$  into Equation 19 yields

$$85 = 168.3 - (3.625) (80) + (0.0555) (80)^2 - (0.0002923) (80)^3 - (0.279) (45) \\ + (24.4) (0.716)^{C/20} + (159.96) (0.333) - (577.191) (0.333)^2 \\ + (487.696) (0.333)^3$$

simplifying

$$24.4(0.716)^{C/20} = 6.46$$

or

$$(0.716)^{C/20} = 0.265$$

from which

$$C/20 = 4$$

therefore,

$$C = 80 \text{ cfm}$$

*Example 2:* It is desired to set an automatic valve-shifting timer on a dehumidification machine so that the cycle is switched to regeneration at a point about halfway between breakpoint and completion times. Inlet air is at 80 F dry bulb and 30 percent RH. Silica gel (6-16 mesh, 42 lb/cu ft dry density) is the adsorbent, and the bed dimensions are 16 in. diameter, 5 in. depth. Rate of air flow is 70 cfm.

Estimate the time after the beginning of adsorption for which the timer should be set.

*Solution:* Refer to Equation 17. Substitution of  $T = 80$ ,  $R = 30$  and  $C = 70$  into Equation 17 yields

$$t_c/G = 83.7 - (2.914) (80) + (0.03143) (80)^2 - (0.0001148) (80)^3 \\ + (0.2799) (30) + (127.63) (70)^{-0.623} \\ t_c/G = 10.32 \text{ min/lb gel}$$

$$\text{Bed volume} = (\text{Face area}) (\text{Depth}) = (\pi) (8/12)^2 (5/12) = 0.5818 \text{ cu ft}$$

$$\text{Dry weight of gel} = (\text{Bed volume}) \times (\text{Dry density}) = (0.5818) (42) = 24.44 \text{ lb}$$

Therefore, *completion time*,  $t_o = (10.32) (24.44) = 252 \text{ min}$

Next, refer to Equation 18. Substituting  $T = 80$ ,  $R = 30$  and  $C = 70$  into Equation 18 yields

$$\begin{aligned} t_B/G &= 4.83 - (0.01836) (80) + (2.781) (0.707)^2 - (0.0588) (70) \\ &\quad + (0.0002968) (70)^2 - (0.0000004952) (70)^3 \\ t_B/G &= 1.51 \text{ min/lb gel} \end{aligned}$$

Therefore, *breakpoint time*,  $t_B = (1.51) (24.44) = 37 \text{ min}$ .

Thus the timer should be set for  $37 + (1/2) (252 - 37) = 144 \text{ min}$ .

#### VIRTUES AND FAULTS OF METHOD

If the experimental results are reasonably accurate and if the functions of the design or prediction characteristics are not unduly physically erratic, the foregoing analysis method has the following virtues: (a) No simplifying assumptions about the physical system as such are made, for example, no particular mechanism of localized adsorption was assumed; the adsorbent particles were not assumed to be smooth spheres; no perfect fluid qualities were ascribed to the adsorbate; (b) It is adaptable to any dynamic system providing the prescribed test methods are used; (c) Based on relatively few experimental tests, it predicts an estimate of those factors which the engineer wants most concerning dynamic adsorbent behavior under a great number of combinations of inlet conditions; (d) Having once formulated the equations for a given system over the desired range and tabulated the results, the method can be readily used by anyone whether or not he is mathematically or technically trained.

The disadvantages inherent to the method are: (a) predicted characteristics at best can be only as accurate as the test data; (b) reliability of predictions decreases toward the range limits of the variables, *i.e.*, as the variables move away from their respective standard values; (c) inter-variability of primary variables is not fully accounted for; (d) each different system requires an experimental investigation and analysis; and (e) certain factors such as adsorbent grain size and bed geometry, which have some effect on adsorbent characteristics, are not considered, although with the added complication of extra tests and extra terms in the equations they could be included.

#### DISCUSSION

GUNNAR C. F. ASKER, Falls Church, Va., (WRITTEN): The authors have pointed out that prediction of dynamic adsorption is complex and influenced by many variables.

The proposed analysis method is adaptable to behavior prediction of solid sorption dehumidifiers with stationary beds with bed depths of 2 in. to 8 in. and air flows used in good design practices. Table 1 of the paper gives the nominal values of test variables. It would have been of value if more tests could have been the basis for the curves used and if bed depths of  $1/2$  in. and 20 in. for example could have been tested also, which would give a more complete Fig. 5 in the extreme conditions. Equation 3 of the paper could in the range 25 to 120 fpm be a constant. Equation 4 could in the range 2 in. to 8 in. bed depth be a constant and these equations might become of higher order than 1 if extreme conditions are included.

If we limit the adaptability of the test method to equipment with a bed depth of 2 in. to 8 in. and stationary beds, the prediction formulas can be simplified by using constants for Equations 3 and 4 without significant sacrifice of accuracy.

The paper clearly emphasizes the complex nature of dynamic adsorption and the need for further research of rate controlling factors.

The authors should be complimented for their mathematical approach to behavior prediction for solid sorption dehumidifiers.

JOHN EVERETTS, JR., Philadelphia, Pa.: In this paper there is a serious error of omission which is not due to the authors but due to the Society. This paper was sponsored thru the efforts of the TAC on Sorbents and I would like to have credit given to the TAC before the paper is finally published.

The paper is one that is of interest from the standpoint of TAC operation, which is made up primarily of manufacturers and people who are interested in sorbents and who decided they needed some basic fundamental information on sorbent materials. They came to the Committee on Research for advice as to how they could get this information, and in the course of our natural operation we advised them that they should work up a program, present it to us, and also present us an idea of how to obtain the funds and where this operation was to be conducted.

They came to us with a very definite concise program; they came to us with one hundred percent of funds contributed by their own industry, and thru the cooperation of The Pennsylvania State University, which has a laboratory which could undertake this research, they went thru the program and came up with the results as indicated in this paper, which is the first that we term fundamental directed research to give some basic data that are of value in design.

G. L. SIMPSON, Pittsburgh, Pa.: The Committee on Sorption is very well pleased with the work of Dr. Ross and Professor McLaughlin, and met at noon today to discuss further work.

An engineer who has been designing adsorption equipment for a long period made the comment to me the other day that there is probably little in this paper that is not known by the engineers and the manufacturing concerns in that industry.

Now whether that is so or not, it does not in any way detract from the value of this paper. That man, when he first started in designing, had no information whatsoever to guide him. He had a little rule-of-thumb furnished by one of the manufacturers of desiccants—*use 10 cubic feet per hour per pound*. That's all he knew to start in his designing.

The early engineers had to compel adsorption to do enough useful work so they could earn a living.

Papers of this kind and this paper in particular strengthen the hand of these engineers. They very greatly contribute to the information for engineers just entering that industry, the new blood, and also contribute to an increased understanding of what we are doing. That will add markets.

There was a year, some twenty years ago, when to the best of my knowledge and belief, the total gross sales of adsorption equipment in this country amounted to \$1,850.00. There were at that time 2 manufacturers of desiccants, one of them so thoroughly discouraged that he was about ready to quit manufacturing. Today there are 4 manufacturers of activated alumina in the United States alone, and a number of manufacturers of silica gel, and there are several new desiccants coming on the market, and the total business is running into the millions of dollars a year.

AUTHORS' CLOSURE (Professor McLaughlin): The request by Mr. Asker for data on  $\frac{1}{2}$  in. and 20 in. thick beds is understandable and perhaps indicates the direction which future work should take. Tests on thicker beds present no special problems. However, to study the activity in beds less than 2 in. thick will require rapid response recorders. Temperature and humidity changes at the bed outlet will occur at a rate beyond the ability of an observer to make manual observations. Virtually continuous records will also be necessary to study the short cycles which will accompany thin beds.



**1542**

## SEMI-ANNUAL MEETING, 1955

SAN FRANCISCO, CALIFORNIA

Meeting at San Francisco for the first time since 1941, the Society held the 1955 Semi-Annual Meeting at the St. Francis Hotel, June 27-29, attracting a total attendance of 662. Final figures show that 335 members, 179 ladies, 123 guests and 25 children were present.

Following the opening of the first technical session in the Italian Room of the St. Francis Hotel, at 9:30 a.m., June 27, Pres. John E. Haines, Minneapolis, Minn., presented D. E. McLeod, president of the Golden Gate Chapter, who expressed his own pleasure, and that of the chapter, on being able to welcome the Society to San Francisco after the 14 years which have elapsed since the 1941 meeting.

President Haines announced that the next order of business would be the presentation of the proposed By-Laws. He read the section of the By-Laws outlining the method of amending them and explained that the proposed By-Laws were presented only at this session and for discussion between this Meeting and the Annual Meeting in Cincinnati in January 1956. He indicated that it was hoped that members would take advantage of the period to discuss these proposed changes and be prepared to vote on them at the time of the Annual Meeting. He then called on Second Vice Pres. P. B. Gordon, New York, N. Y., to present the changes.

Vice President Gordon presented to the meeting a written notice of proposed amendments to the Society By-Laws, and which notice is summarized in the following paragraphs.

That *Section 1 of ARTICLE V on The Council* be amended by adding to the first sentence the phrase "seven (7) of whom shall be from different Regional Areas and elected as Regional Directors for their respective Areas" and that *Section 2* be amended by inserting the words "Regional Areas" between the words "Branches" and "Officers" in Line 7 of the present By-Laws.

Amendments to **ARTICLE VII on Committees, Section 2**, include a prefix reading as follows: "Unless otherwise provided, the" etc. and that a new subdivision (e) be added to *Section 2*, establishing a Regions Central Committee, consisting of the Second Vice-President and the seven (7) Regional Directors.

That *Section 3 (a)* describing the duties of the Admission and Advancement Committee be amended.

That *Section 3 (c)* be amended to permit the GUIDE Committee to serve for a term of one (1) year commencing November 1.

That *Section 3 (f)* be amended to provide for seven (7) Chapters Regional Committees, each serving one Regional Area, and each consisting of the Regional Director for the Area and (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year.

That *Section 3 (g)* be eliminated, also *Rules 1 and 2*, and that *Section 5 (c)* be amended by adding a prefix thereto reading as follows: "Except as otherwise provided,".

In **ARTICLE II, Memberships, Sections 1, 2 and 3(a)** to be amended and **3(d)** be added to to provide for a new grade of Fellow and to eliminate the Junior Member grade. Also that **Section 3(d)** be made **3(e)**, that present **3(e)** be **3(f)** and **Section 3(h)** be amended. That **Sections 4, 5 and 6** be amended outlining the privileges, limitations and procedures for proposals and applications and prerequisites for the various membership grades. Also that the opening phrase of **Section 8(a)** be amended to include Fellows and eliminate reference to Junior grade.

Amendments to **ARTICLE III on Chapters, Special and Student Branches, and Regional Areas** provide for renumbering existing **Sections 1, 4 and 5** as **Sections 1, 2 and 3**. That a new **Section 4** on regional areas be added and that existing **Sections 2 and 3** be renumbered as **Sections 5, 6** and present **Section 6** be renumbered **Section 7**.

In **ARTICLE IV on Funds**, use the word "Junior" for "Associate" in **Section 1**. Amend **Section 3** on **Dues** to continue exemption for Honorary, Presidential and Life Members and annual dues rates as follows: Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates, twenty-five (\$25.00) dollars. The annual dues of Associate Members under thirty (30) years of age shall be fifteen (\$15.00) dollars. The annual dues of Students shall be fixed by the Council and shall be published in the JOURNAL.

That **Section 5** on **Allocation of Dues for Research**, be amended to provide that forty (40%) percent of the dues received from Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates, be allocated for basic or fundamental research.

**ARTICLE VIII on Meetings, Nominations, and Elections** provides for an amendment to **Section 3** on the make-up of the Nominating Committee members and alternates selected by Council and the Chapters Regional Committees, and a requirement for meetings at the Annual and Semi-Annual Meetings of the Society.

An amendment that **APPENDIX A** of the By-Laws be eliminated.

Vice President Gordon outlined the meaning of the several effects these amendments could be expected to have. He also stated that the amendments were proposed by the Council and were endorsed by more than two-thirds of the Council members.

President Haines then appointed the following to serve as the Committee on Resolutions: John Everetts, Jr., Philadelphia, Pa., Chairman; G. C. F. Asker, Washington, D. C., B. L. Evans, St. Louis, Mo., and D. M. Mills, Houston, Tex.

President Haines then turned the gavel over to H. B. Nottage, vice chairman of the first technical session and three papers (see program on p. 336) were presented and discussed.

At the second technical session on Tuesday, June 28, 9:30 a.m., Vice-Pres. John W. James presented vice chairman B. H. Jennings who presided over the technical session during which four papers were presented and discussed, one being by title only.

At the third technical session, Wednesday, June 29, 9:30 a.m., Second Vice-Pres. P. B. Gordon called on vice chairman T. E. Taylor who presided at the session during which four technical papers were presented and discussed.

At 2:00 p.m. on Wednesday, June 29, the Evaporative Cooling Symposium opened, being presided over by Treas. E. R. Queer. The panel consisted of: Wm. T. Smith, Chairman, Chief, Refrigeration and Air Conditioning Section, Headquarters, U. S. Air Force, Washington, D. C.; R. E. Phillips, Jr., Vice President, Ralph E. Phillips, Inc., Los Angeles, Calif.; S. F. Duncan, Director of Research, Farr Co., Los Angeles, Calif.; R. J. Petersen, Chief Engineer, Utility Appliance Corp., Los Angeles, Calif.; Stuart Giles, Director, Heat and Sanitation

Division, U. S. Naval Civil Engineering Laboratory, Port Hueneme, Calif.; Robert Ash, Assistant to the President, International Metal Products Co., Phoenix, Ariz.; Richard Hukill, Supervisory Mechanical Engineer, U. S. Naval Civil Engineering Laboratory, Port Hueneme, Calif.; D. T. Robbins, Chief Design Engineer, Holmes & Narver, Inc., Los Angeles, Calif.; R. M. Westcott, Partner, Holladay & Westcott, Los Angeles, Calif.

Each member of the panel presented his prepared remarks and discussion from the floor followed. A condensed record of the technical material presented at this Symposium appears in the JOURNAL SECTION of *Heating, Piping and Air Conditioning*, August, 1955, pages 141-147, inclusive.

Following the completion of the Symposium, President Haines called on John Everetts, Jr. for the Report of the Committee on Resolutions which was presented as follows:

WHEREAS, the first Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS is now about to close; and

WHEREAS, the Meeting was held in the beautiful city of San Francisco famed for its hills, valleys and natural air conditioning and where the weatherman has fully cooperated with the aims and ideals of our Society; and

WHEREAS, under the excellent direction of our officers and staff, the aims of the Society have been advanced through the presentation of technical papers and discussions; and

WHEREAS, the Golden Gate Chapter through its Committee on Arrangements of which G. M. Simonson is Honorary Chairman, S. W. Terry, Sr., General Chairman, Herbert L. Duncan, Vice Chairman and Donald E. McLeod, Chapter President as well as the members, have ably provided an excellent program of entertainment of diversified nature for the enjoyment of their guests and providing visitors with an educational guide entitled "Sin in San Francisco"; and

WHEREAS, it has been very difficult for many of the bachelor members to differentiate the smog from the fog because of the grog

BE IT RESOLVED, that, as the Semi-Annual Meeting with the greatest all time attendance of 662 members and guests, we express our gratitude and appreciation to the Golden Gate Chapter, its officers, committees and members for their outstanding work, and

TO Major General Wm. F. Dean, U. S. A. for his interesting and educational talk at the welcome luncheon, and

TO the hotel management and staff for their courtesy and promptness,

TO the authors and the discussors of technical papers for their educational contribution,

TO Walter Swanson, Executive Vice President of the San Francisco Convention and Visitors Bureau, for his fine cooperation and the assistance of his staff in helping to make this an outstanding meeting,

TO the San Francisco newspapers, the wire services and the trade paper representatives for their excellent support and publicity,

TO the ladies of San Francisco for the complete and unselfish attention they have given to the visiting ladies,

TO Dr. Baldwin M. Woods, Society Past President, for his usual masterful job as toastmaster,

TO Reverend John Collins for his invocation at the Banquet,

TO Dr. A. M. Zarem for a most enlightening and interesting talk on an important problem,

TO President John E. Haines, the Officers and Committees of the Society for their tremendous contribution in time and energy for the benefit of the Society in this, our 62nd year,

Respectfully submitted,

John Everetts, Jr., *Chairman*  
Gunnar C. F. Asker  
Bruce L. Evans  
D. M. Mills

Following the unanimous adoption of these resolutions, the meeting was adjourned.

## PROGRAM—SEMI-ANNUAL MEETING

St. Francis Hotel, San Francisco, Calif.—June 27–29, 1955

### Friday, June 24

8:00 p.m. Executive Committee, John W. James, *Chairman* (Room 218)

### Saturday, June 25

1:30 p.m. Finance Committee, E. R. Queer, *Chairman* (Room 212)

3:00 p.m. Chapter Relations Committee, P. B. Gordon, *Chairman* (Room 218)

### Sunday, June 26

9:30 a.m. Council Meeting, (Room 220)

10:00 a.m. REGISTRATION—(Green Room)

2:00 p.m. Research Executive Committee, B. H. Jennings, *Chairman* (Room 212)

3:00 p.m. WELCOME TEA, Sacramento Chapter Hosts

### Monday, June 27

9:00 a.m. REGISTRATION—(Green Room)

9:30 a.m. FIRST TECHNICAL SESSION, (Italian Room)

Call to Order by Pres. John E. Haines

Welcome by Don McLeod, President, Golden Gate Chapter

Amendments to By-Laws

1. Membership Grades and Qualifications

2. Regional-Director Plan for Chapter Operation

3. Guide Committee Appointment

4. Student Application Processing

H. B. Nottage, *Vice Chairman*

Electric Analogue Prediction of the Thermal Behavior of an Inhabitable Enclosure, by Harry Buchberg, Los Angeles, Calif., presented by Professor Buchberg.

A Method for Determining Winter Design Temperatures, by M. K. Thomas, Toronto, Ont., Canada, presented by Mr. G. O. Handegord, Saskatoon, Sask.

Periodic Heat Flow Through Flat Roofs, by D. J. Vild, Cleveland, Ohio, M. L. Erickson, Minneapolis, Minn., G. V. Parmelee and A. N. Cerny, Cleveland, Ohio, presented by Mr. Vild.

- 12:30 p.m. WELCOME LUNCHEON, (*Colonial Room*)  
*Chairman:* G. M. Simonson  
*Toastmaster:* C. E. Bentley  
*Speaker:* Major General William F. Dean, U.S.A.  
*Subject:* *Military Requirements of Today*
- 1:00 p.m. GOLF TOURNAMENT, *Olympic Country Club*
- 2:00 p.m. Long Range Research Program, John Everetts, Jr., *Chairman*, (*Room 212*)
- 2:00 p.m. Membership Committee, B. W. Farnes, *Chairman*
- 2:00 p.m. TAC on Air Cleaning, A. B. Algren, *Chairman*, (*Room 220*)
- 2:00 p.m. SIGHTSEEING TRIP—Scenic Boat Ride on San Francisco Bay
- 7:00 p.m. "FORTY-NINER" GET-TOGETHER PARTY, *Surf Club* at Beach

### Tuesday, June 28

- 9:00 a.m. REGISTRATION—(*Green Room*)
- 9:30 a.m. SECOND TECHNICAL SESSION, (*Colonial Room*)  
 Call to Order by First Vice Pres. John W. James  
 B. H. Jennings, *Vice-Chairman*  
 Ventilation of Commercial Laundries, by Sidney Marlow, New York, N. Y., presented by E. R. Kaiser, Cleveland, Ohio.  
 Air Conditioning Coil Odors, by A. B. Hubbard, Bloomfield, N. J., Nicholas Deininger and Frederick Sullivan, Cambridge, Mass., presented by Mr. Hubbard.  
 A Rapid General Purpose Centrifuge Sedimentation Method for Measurement of Size Distribution of Small Particles, Part II—Procedures and Applications, by K. T. Whitby, presented by the author.  
 Size Distribution and Concentration of Airborne Dust, by K. T. Whitby, A. B. Algren and R. C. Jordan, Minneapolis, Minn., presented by Dr. Whitby.
- 10:30 a.m. SIGHTSEEING TRIP: Golden Gate Bridge, Muir Woods
- 1:30 p.m. Nominating Committee, B. L. Evans, *Chairman*, (*Room 221*)
- 2:00 p.m. Chapters Conference Committee, J. S. Burke, *Chairman*, (*Room 261*)
- 2:00 p.m. Public Relations Committee, J. H. Fox, *Chairman*
- 6:00 p.m. INFORMAL DINING AT FISHERMAN'S WHARF, International Settlement
- 8:45 p.m. SAN FRANCISCO NIGHT CLUB TOUR

### Wednesday, June 29

- 9:00 a.m. REGISTRATION—(*Green Room*)
- 9:15 a.m. Ladies' Continental Breakfast, (*Mural Room*) Conducted Tour of Stores, Luncheon in Chinatown
- 9:30 a.m. THIRD TECHNICAL SESSION, (*Colonial Room*)  
 Call to Order by Second Vice Pres. P. B. Gordon  
 T. E. Taylor, *Vice-Chairman*  
 Sources of Vent Gas in Hot Water Heating System, by L. N. Montgomery, Boston, Mass., and W. S. Harris, Urbana, Ill., presented by Professor Harris.  
 Psychrometric Analysis for Design of Forced Draft Cooling Towers, by S. E. Agnon, Haifa, Israel, and B. H. Spurlock, Jr., Boulder, Colo., presented by Professor Spurlock.  
 Resistance of Wooden Louvers to Fluid Flow, by C. W. Bevier, College Station, Tex., presented by Dr. H. B. Nottage, Encino, Calif.  
 Performance and Evaluation of Room Air Distribution Systems, by Alfred Koestel and G. L. Tuve, Cleveland, Ohio, presented by Professor Tuve.

- 10:00 a.m. Ladies' and Children's Sightseeing and Luncheon: Golden Gate Park, Aquarium, Zoo and Ocean
- 2:00 p.m. EVAPORATIVE COOLING SYMPOSIUM—(*Colonial Room*)  
 Call to Order by Treasurer E. R. Queer  
 Wm. T. Smith, *Chairman*  
 Historical—R. E. Phillips, Jr.  
 Air Cooling by Evaporation—S. F. Duncan  
 Evaporation from Surfaces—R. J. Petersen  
 Weather Data Limitations—Stuart Giles  
 Geographical Limitations—Robert Ash  
 System Design—Richard Hukill  
 Indirect Systems—D. T. Robbins  
 Water Treatment—R. M. Westcott  
 Discussion Period  
 Report of Committee on Resolutions  
 Unfinished Business  
 New Business  
 Adjournment
- 6:30 p.m. SOCIAL HOUR—(*Italian Room*)
- 7:00 p.m. SEMI-ANNUAL BANQUET—(*Colonial Room*)  
*Toastmaster:* Dr. Baldwin M. Woods, ASHAE Past President; Vice President, University Extension, University of California  
*Speaker:* Dr. A. M. Zarem, Manager, Southern California Division, Stanford Research Institute  
*Subject:* Smog—A Challenge to Technology

### COMMITTEE ON ARRANGEMENTS

G. M. SIMONSON, *Honorary Chairman*

S. W. TERRY, *General Chairman*

H. L. DUNCAN, *Vice Chairman*

*Banquet Committee:* N. H. Peterson, *Chairman*, J. E. Murray, T. R. Simonson.

*Entertainment Committee:* J. I. Sprott, *Chairman*, Edward Hill, Jr., W. M. Lewis, J. E. Murray, C. W. Reid, Lawrence Shea.

*Finance Committee:* K. F. Baldwin, Jr., *Chairman*, Wm. Barron, J. D. Kniveton, J. O. Martin, J. N. Moore, T. R. Simonson.

*Ladies Committee:* E. C. Cooley, Jr., *Chairman*, A. T. Bloxham, T. E. Brewer, Charles Lambert, Richard Stites, Jr.

*Publicity Committee:* T. E. Brewer, *Chairman*, A. W. Berrier, W. H. Buck, R. E. Conner.

*Reception Committee:* T. J. White,

*Chairman*, G. N. Aranovsky, R. D. Knowles, M. J. Kodmur, W. W. MacLean, F. H. McLaughlin, J. D. Mitchell, H. D. Page, R. M. Scott, G. H. Smith, J. B. Smith.

*Sessions Committee:* D. L. Williams, *Chairman*, L. Britt, Leo Dwyer, G. L. Gendler, R. B. Holland, F. W. Jordan, R. C. Kelly, R. N. Poage, E. C. Sanford, C. B. Smith, J. E. Wilson.

*Sports Committee:* Dudley Deane, *Chairman*, J. C. Beck, Jr., D. A. Delaney.

*Transportation Committee:* K. K. LaPoint, *Chairman*, P. S. Beggs, A. H. Blackwell, A. T. Bloxham, H. A. Wolfson.

*Young People's Committee:* R. C. Pri-buss, *Chairman*, J. C. Beck, T. C. Douglass, Jr., J. F. McIndoe.



**1543**

## ELECTRIC ANALOGUE PREDICTION OF THE THERMAL BEHAVIOR OF AN INHABITABLE ENCLOSURE†

By HARRY BUCHBERG\*, LOS ANGELES, CALIF.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with the Department of Engineering of the University of California.

THIS PAPER presents a thermal circuit analysis of the thermal response of a simple dwelling to the time variable micro-climate and shows how solutions of the circuit are obtained by electrical analogy. Consideration was given only to the temperature-time response of the structure and space and to the rate of heat loss or gain by the space without including moisture transfer. A direct determination of the sensible air conditioning load required to maintain the space at a constant temperature was also made.

Methods of calculating the instantaneous heat load based on solutions of the heat conduction equation are available in the literature. Under many conditions calculations based on these methods result in substantial errors of prediction. This is especially true when the influence of radiation exchange within the enclosure, direct solar transimssion through transparent areas, ventilation, and thermal capacity of internal partitions and objects are of importance. Many experimental studies of the thermal behavior of full-scale systems have been undertaken and can be found in the literature. Such studies are costly in time and money and it behooves us to seek and develop methods of analysis which consider the entire system at once including the interactions at the boundaries.

One of the most powerful techniques for predicting the thermal behavior of complex heat transfer systems is the representation of the system and boundary conditions by a thermal circuit. Many investigations using this method of analysis have been reported in the literature<sup>1</sup>. Examples of the application of thermal circuit techniques to the specific study of the thermal behavior of buildings are few. Important contributions occur in References 2, 3, 4, 5 and 6.

Space limitation made necessary the omission of considerable detail concerned with the representation of boundary conditions, the calculation of circuit parameters, the radiation exchange network within the house, and methods of lumping

† The investigation was carried out by the author in partial satisfaction of the requirements for the degree of Master of Science in Engineering.

\* Department of Engineering, University of California.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, Calif., June 1955.

<sup>1</sup> Exponent numerals refer to References.

circuit parameters. Information on these subjects is given in Appendixes A, B, and C and in a second paper, Electric Analogue Studies of Single Wall Sections, soon to be available.

#### SYSTEM INVESTIGATED

*General Description:* A one-room wood frame dwelling, shown in Fig. 1, constituted the thermal system studied. It is referred to as the *test house*. The test house is located on the roof of the Engineering Building B at the University of California, Los Angeles. This places the house approximately 50 ft above the



FIG. 1. TEST HOUSE LOCATED ON ROOF OF ENGINEERING BUILDING, UNIVERSITY OF CALIFORNIA, LOS ANGELES

ground level (longitude 118 deg 26 min, latitude 34 deg 4 min N) with a substantially unobstructed view to the east, west and south. The north side is obstructed by a 7-ft parapet running parallel to the north facing wall 10 ft distant. The long walls face exactly north and south, the south facing windows being protected from direct solar radiation during the months from June to October by eaves overhang. The south wall windows constitute the only glass area. The roof is very nearly a horizontal plane and is so considered in the analysis. As shown on Fig. 2, a schematic drawing of the thermal system being considered, the interior dimensions are 10.4 x 15.4 x 8.2 ft.

*Construction Details:* The test house is of Douglas Fir wood frame construction with walls of 1 in. exterior fir sheathing, 2 x 4 in. vertical studs placed 16 in. on center, and 3/4 in. interior plywood. The west wall only contains rockwool insulation batts (nominal 3 in. thick) in the air space between the studs. The ceiling is of open joist construction and the floor consists of 1 in. fir flooring nailed to 2 x 4 in. joists which rest on leveling wedges and on the roof of the Engineering Building. The roof covering is green roll asphalt roofing, one section of which is painted flat white as indicated in Fig. 2. The south wall windows are made of standard single glass sheets set into integral wall framing. Exterior surfaces are all painted a buff color.

*Boundary Conditions during Test Period:* The thermal behavior of the test house depends on the geometry and on the thermal properties of the structure and also

upon independent functions or boundary conditions that describe the interaction of the system with its surroundings.

During the test period (week of September 7, 1953) the days and nights were clear and air temperature and movement were normal for that time of year except that the peak air temperature occurred later than usual. Radiation exchange occurred between the test house and the sun, the air mass, the engineering building roof, and the parapet at the north wall. Air movement was somewhat restricted under the floor of the house. A significant heat source existed inside the test house due to the recording equipment located there.

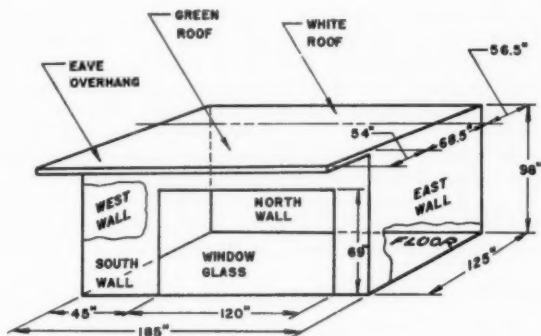


FIG. 2. SCHEMATIC DIAGRAM AND DIMENSIONS OF TEST HOUSE

**Instrumentation:** The test house was provided with instrumentation for measuring the physical variables of the system that have an important influence on the thermal behavior. Measurements were made of the following variables:

1. Inside and outside surface temperatures.
2. Inside and outside wet and dry-bulb air temperatures.
3. Heat fluxes across several inside surfaces.
4. Inside and outside air velocity.
5. Solar and total irradiation of a horizontal surface.

Conventional instrumentation such as thermocouples, aspirated psychrometers, heat meters, a hot coil anemometer, an Eppley pyrlieliometer, and a total hemispherical radiometer<sup>7</sup> was used for all observations. Measurements were recorded continuously during the test period by high-speed, electronic, self-balancing potentiometers and a d-c amplifier and recording oscillograph, and provided the basis for the experimental results presented later in this paper.

## THERMAL CIRCUIT PRINCIPLES

Circuit representation of physical systems is a well established technique and requires little elaboration at this point. However, a few statements concerning the circuit representation of thermal systems and this particular application are in order.

*Limitations and Advantages:* At the outset it is important to recognize that the thermal system represented by a circuit is not the real physical system but rather an approximately equivalent system composed of discrete sections or lumps. Associated with each lump is a thermal capacity, all capacitors being appropriately interconnected by thermal resistance linkages, representing resistance to thermal energy transfer. Thus, the real distributed property system is replaced by a system of pure lumped thermal capacity interconnected with pure lumped thermal resistance. The 2 systems may be thought of as approaching equivalence as the size of the lumps become smaller and as the number of lumps approach infinity.

The error introduced by lumping the properties of the system is a function of the fineness of the lumping, the pattern of the lumping, the boundary conditions, the length of time after a change in input occurs, and the location in the network where the value of temperature or heat flux is required. An exact quantitative evaluation of the error can be made only by comparing the exact analytical solution of the distributed system with the solution of the difference equations representing the lumped properties system.<sup>8</sup> This procedure becomes very involved even with relatively simple boundary conditions and prohibitive with the boundary conditions involved in this investigation. The question of lumping was, therefore, evaluated experimentally by comparing the response of several different analogous electrical networks representing a single wall section. It is planned to present a report on these studies in the later paper, Electric Analogue Studies of Single Wall Sections.

#### THERMAL CIRCUIT REPRESENTING THE TEST HOUSE

In order to work out a suitable thermal circuit for the test house, it was necessary to idealize the real physical system in the following:

1. All thermal energy transfer through structural elements is unidirectional and perpendicular to the long dimension of the element.

2. The test house is considered to be made up of 8 structural elements consisting of the west wall, east wall, north wall, south wall, south glass area, floor, white roof and green roof; all of which present parallel paths of thermal energy transfer into the enclosed space.

3. Each of the distributed property sections is replaced by lumped property sections.

4. The lumped thermal properties are constant and equal to the mean values of the real properties over the temperature range encountered.

5. The instantaneous temperature representing the average temperature of any plane surface is approximately equal to the actual temperature of the surface measured at its geometric center.

6. The air mass filling the enclosed space is considered to be at a uniform temperature at any instant, *i.e.*, thermal energy transferred to or from the air acts instantaneously on the whole mass to raise or lower its temperature uniformly space-wise.

7. Net radiation exchange with the inside air mass is negligible.

8. All surfaces radiate diffusely.

9. Negligible infiltration and leakage exists.

10. Any existing moisture transfer is negligible.

11. Incident solar radiation transmitted by any section is negligible.

12. Contact resistances through the conduction path are negligible.

As a result, the diagram, Fig. 3, shows the thermal circuit of what may be called the *idealized* test house, and is the circuit which was solved to obtain the predictions mentioned later. (The symbols used in Fig. 3 are defined in Nomenclature and Symbols:). For purposes of description the circuit may be thought of as consisting of 3 parts:

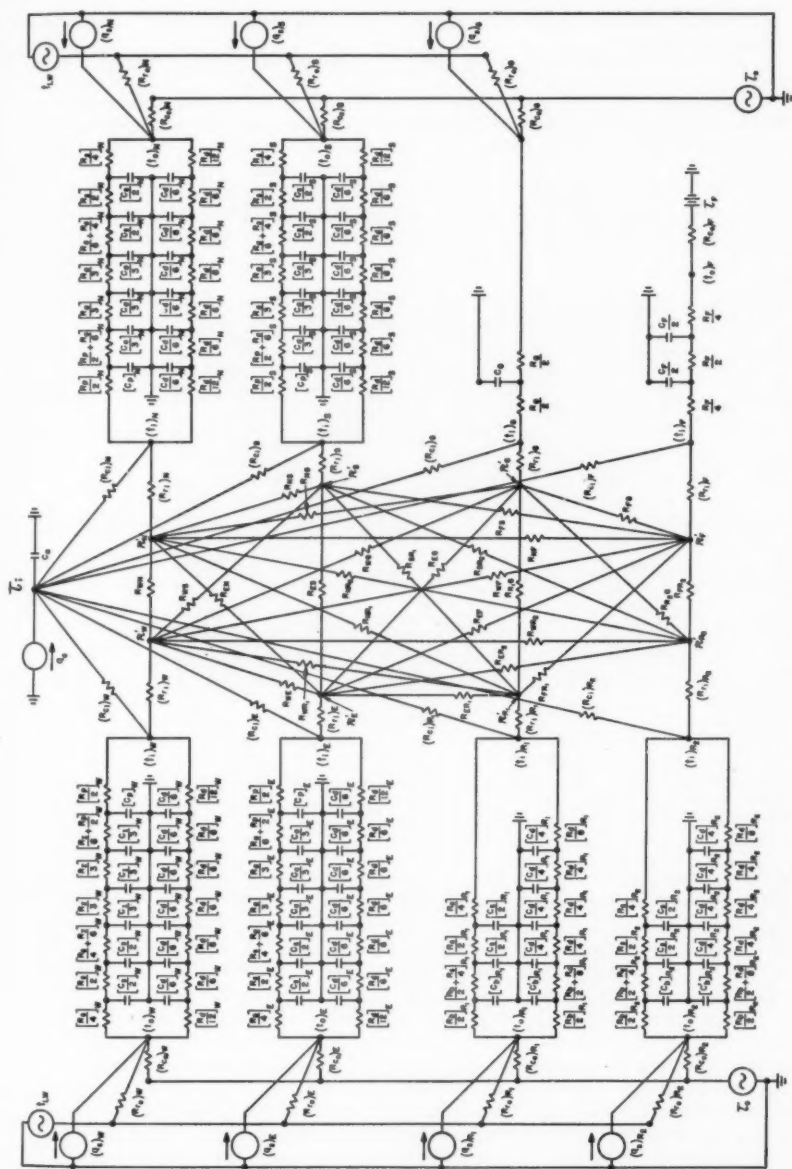


FIG. 3. THERMAL CIRCUIT REPRESENTING THE IDEALIZED TEST HOUSE

- ✓ 1. Thermal conduction paths through the structural elements represented as parallel lumped R-C networks.
- ✓ 2. Radiation exchange between all interior surfaces represented by a resistance network.
- ✓ 3. Boundary conditions represented by time variable and constant temperature sources connected through appropriate thermal resistance to the network, and thermal energy generators at various points in the network.

On the inside of the test house thermal energy exchange takes place by convection to the air mass and by radiation exchange between all of the surfaces. The convection exchange is simply treated as a resistance to air mass, represented by

TABLE 1—SUMMARY OF RATIOS AND UNITS OF ANALOGOUS ELECTRICAL AND THERMAL QUALITIES<sup>a</sup>

QUANTITY	UNITS		SCALE FACTORS	
	THERMAL	ELECTRICAL	RATIO	VALUE
Time	hrs	sec	$\frac{\theta_e}{\theta_t}$	2
Capacity	$\frac{\text{Btu}}{^{\circ}\text{F}}$	Farads	$\frac{C_t}{C_e}$	$8 \times 10^5$
Resistance	$\frac{^{\circ}\text{F}}{(\text{Btu/hr})}$	Ohms	$\frac{R_e}{R_t}$	$16 \times 10^5$
Potential	$^{\circ}\text{F}$	Volts	$\frac{E}{i - i^{*b}}$	1
Rate of Energy Transfer	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{Coulombs}}{\text{sec}}$ or Amperes	$\frac{q}{i}$	$16 \times 10^6$

<sup>a</sup> Refer to Appendix B for the actual values of resistances and capacitances in thermal and electrical units used in the network representation.

<sup>b</sup>  $i^* = \text{Ref. Temp.}$

a thermal capacitor. The radiation exchange is represented by a linearized resistance network that accounts for all interreflections as well as the first or fundamental exchange. It is recognized that interreflections are not of great importance in this instance, all emissivities being near unity; however, with little additional complexity all interreflections can be handled (see Appendix C). This serves as an example for cases where interreflections may be of importance.

The presence of recording equipment resulted in a significant thermal energy source within the enclosed space. From the instrument manufacturers ratings, it was estimated to be approximately 375 watts and is represented in the thermal circuit as a constant energy source or input device connected to the air capacitor.

To determine the sensible air conditioning load required to maintain the space at a constant temperature, a constant potential source was connected to the inside air temperature point and then grounded through an appropriate resistance. The potential drop across the resistance is proportional to the load.

## THERMAL CIRCUIT SOLUTION BY ELECTRICAL ANALOGY

*Analogous Electric Circuit:* The existence of the thermal-electrical analogy can easily be verified by comparing the finite difference equations representing the lumped thermal circuit with the equations representing a similar electrical circuit. This has been done many times and will not be repeated here. Suffice it to say that the analogy exists for any complex resistance-capacitance network providing the following conditions are met: (a) electrical and thermal circuits are schematically similar; (b) dimensionless moduli characterizing the electrical circuit are numerically equal to those characterizing the thermal circuit; (c) boundary conditions imposed are exactly the same for the electrical and thermal circuits and are correctly related in time.

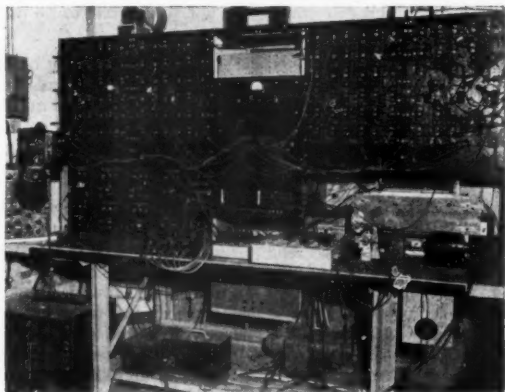


FIG. 4. ELECTRIC ANALOGUE OF TEST HOUSE THERMAL SYSTEM INCLUDING (a) INPUT DEVICES, (b) R-C NETWORK, AND (c) MEASURING INSTRUMENTS

To assure exact similarity between the electrical circuit used and the thermal circuit representing the test house the circuit diagram shown in Fig. 3 was considered to be the analogous electrical circuit. Appropriate scale factors derived to satisfy condition (b) are shown in Table 1.

*Boundary Conditions:* To duplicate the boundary conditions of the thermal circuit it was necessary to provide electrical devices capable of producing the appropriate time-variable potentials, constant potential, and time-variable current inputs. Fig. 4 shows how this electrical equipment appeared when erected. In addition, it was necessary to establish the proper timing between the various inputs.

1. The time-variable potential corresponding to the *diurnal variation in air temperature* was obtained by means of a drum-type function generator. The particular function to be generated was plotted on rectangular coordinate paper. A copper or brass wire was laid over the curve and cemented to the paper which was then formed around a drum and secured. The drum was rotated by a drive mechanism causing the wire curve to act as a wiper against a fixed linear potentiometer. A constant voltage was

impressed across the potentiometer resulting in an output voltage signal at the wire wiper proportional to the air temperature. Zero control and amplification of this signal was accomplished by means of a direct-coupled feed-back amplifier known as an operational amplifier. Amplitude control was obtained by varying the ratio of input to feedback resistance, and zero control was obtained by adding a constant voltage to the function generator output through the operational amplifier. The signal was impressed on the network as a repeating function corresponding to the 24-hr air temperature cycle, and timed with the solar current inputs through a common drive mechanism.

A switching device consisting of timed relays was used to vary the outside convective resistance. It was found that the wind velocity averaged about 5 mph during the day and about 1 mph at night. Therefore, a different value of resistance was switched into the network during the night and day periods. A greater refinement in representing the non-linear convective resistance can readily be obtained by using a stepping switch and many values of resistance corresponding to the pattern of wind velocity during the 24-hr period. This refinement was not considered to be important in this investigation.

2. The long-wave radiation boundary potential was considered to be equal to a constant fraction of the air potential. This time-variable voltage signal was obtained by impressing the function generator output on the network through another operational amplifier used to adjust the amplitude as required.

3. The constant under floor air potential was obtained by means of a regulated d-c power supply.

4. Time-variable current representing the incident solar radiation input absorbed by each of the exposed surfaces was obtained through a special solar input circuit. Equations representing direct and diffuse solar irradiation absorbed by each of the walls were derived (see Appendix A). All of the equations are sine or cosine functions of time. The solar input circuit is simply a device to synthesize these equations and to deliver a time-variable current that is proportional by the proper scale factor to the incident solar radiation absorbed by each surface. The sine and cosine functions were generated by means of Scotch Yoke mechanisms which imparted a sinusoidal motion to the wipers of linear potentiometers. Operational amplifiers were used as adders and for amplitude control. Relays energized through a microswitch actuated by a cam on the Scotch Yoke crank were used to simulate sunrise and sunset. The output was limited to positive values of current by shorting the signal to ground through a diode vacuum tube at appropriate points in the circuit. Currents proportional to the amplifier output voltages were obtained through the use of current generators that produced a current output proportional to the voltage input.

*Methods of Measurement:* Measurement of voltages representing the inside and outside surface temperatures of all sections of the house and the inside air temperature at approximately 20 min intervals during a 24 hr cycle constituted the solution of the thermal circuit. The potentials were recorded on 35 mm film by means of a high speed electronic switch, cathode-ray oscillograph and oscillo-record camera.<sup>9</sup> A high impedance voltmeter was used for spot checks during a run.

In the determination of the sensible air conditioning load a d-c amplifier and recording oscillograph were used to measure the voltage drop across the resistor connected between the battery representing the constant inside air potential and ground.

*Analogue Results:* When the data from these measurements were converted to their thermal equivalents, they were plotted in the accompanying figures as points and labelled *Points-computed*. They are also referred to later as *analogue predictions* when they are compared with the *experimental results*, obtained by the data from instruments provided in the test house.

## EXPERIMENTAL RESULTS

Results of the observations of the thermal behavior of the test house during the 24-hr period beginning at midnight, September 8, 1953, are presented in Figs. 5 through 10. On these figures these results are curves without data points so that computed values representing the electric analogue predictions might be superposed wherever applicable for purposes of comparison.

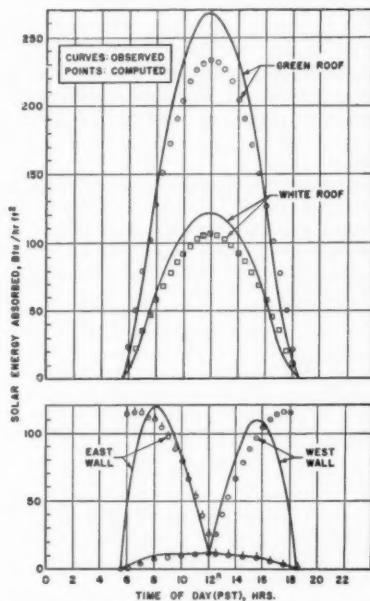


FIG. 5. SOLAR ENERGY ABSORBED DURING THE 24 HR PERIOD BEGINNING SEPTEMBER 8, 1953 (COMPUTED POINTS ARE ANALOGUE INPUTS; CURVES ARE DIRECT MEASUREMENTS)

*Irradiation Data:* The solar energy absorbed by the white and green sections of the roof is shown in Fig. 5. The curves, illustrating the experimental results, are based on the observed irradiation and the same values of absorptivity used in the analogue computations. The solar energy absorbed by the east and west walls is shown in Fig. 5. The observed data illustrated by the curves were calculated from the expression

$$q_a/A_n = (\alpha_{aw})_n [(H_D)_n + \frac{1}{2} (H_d)_o] \quad (1)$$

where

$$(H_D)_n = (H_D)_o [\cos (Z_n - Z_n) / \tan h_n] \quad (2)$$

$(H_D)_0$  and  $(H_A)_0$  being based on the pyrheliometer measurements. Equation 2 is simply arrived at from  $(H_D)_n = (H_D)_N \cos X$  and the definition of  $\cos X$  in the horizon system of coordinates. The value for  $\alpha_{nw}$  was the same as that used in the analogue computations and the azimuth and altitude angles of the sun were obtained from Reference 10.

The net thermal energy lost by the white and green roof sections due to low temperature, long-wave radiation exchange is plotted on Fig. 6. The observed data (the curves) were obtained from the expression

$$(q_r)_{\text{net}}/A_n = (\alpha_{lw})_n H_{sky} - (\epsilon_{lw})_n \sigma T_n^4 \quad (3)$$

where  $H_{sky}$  and  $T_n$  are measured quantities. Values for the radiation properties were the same as those used in the analogue computations.

**Temperature Data:** Several items regarding the temperature data given by the curves in Figs. 5 through 10 are noted as follows:

1. A drop in observed ceiling temperature under the green roof section beginning just before 12 noon may be noted on Fig. 7. According to steady state calculations an increase in conduction path resistance through the roof of approximately 3.5 times could cause a drop in ceiling temperature of about 10 F with the assumption that  $(t_o)_{R2}$  and  $\tau_1$  are substantially independent of the roof conduction resistance. At this time of day  $(t_o)_{R2}$  is influenced mainly by the net radiation exchange and outside convection resistance. An increase in conduction path resistance by a factor of 3.5 can be realized by the presence of an air space approximately  $\frac{1}{2}$  in. thick. The presence of such an air pocket over the measuring point was substantiated by other data which indicate that the degree of dip in the ceiling temperature during this period is very sensitive to the outside surface temperature. On very bright warm days with high peak roof temperatures the dip was accentuated while on cool cloudy days the dip disappeared. This was undoubtedly caused by expansion and contraction of the air pocket (verified by inspection), a well known defect in rolled roof coverings.

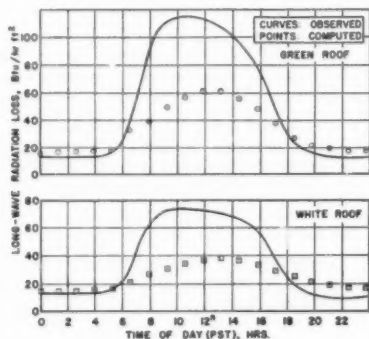


FIG. 6. LONG-WAVE RADIATION LOSS FROM WHITE AND GREEN ROOF SECTIONS DURING THE 24 HR PERIOD BEGINNING SEPTEMBER 8, 1953 (COMPUTED POINTS ARE ANALOGUE INPUTS; CURVES ARE DIRECT MEASUREMENTS)

2. Referring to Fig. 8, it can be seen that the outside surface temperature of the east wall rose very rapidly shortly after 8:00 a.m. On the other hand, the solar radiation absorbed by an unobstructed east wall based on measurements of horizontal irradiation increased steadily after 5:40 a.m. reaching a peak at 8:00 a.m. as shown in Fig. 5. The reason for this seemingly inconsistent behavior can be explained by the fact that the east wall of the test house, at the level of measurement, was actually shaded from the sun until approximately 8:00 a.m. by a building wall approximately 50 ft to the east.

3. Observed south wall surface temperatures are not shown on Fig. 9 because these temperatures were not recorded during the test period.

Wind velocity data as a function of time are not presented in this paper, although measurements were made. Reduction of the oscillogram tapes require more time and effort than was thought to be warranted at this time. The average wind velocity on September 8, 1953 was between 4 and 5 mph during the day and about 1 mph at night.

#### ANALOGUE PREDICTIONS COMPARED WITH EXPERIMENTAL RESULTS

Predictions of the thermal behavior of the test house are shown as points superposed on graphs of observed temperature-time history for a 24-hr cycle during the test period. Figs. 7, 8, and 9 present the surface temperatures, and Fig. 10 the diurnal variation in outside and inside air temperature. The boundary inputs for

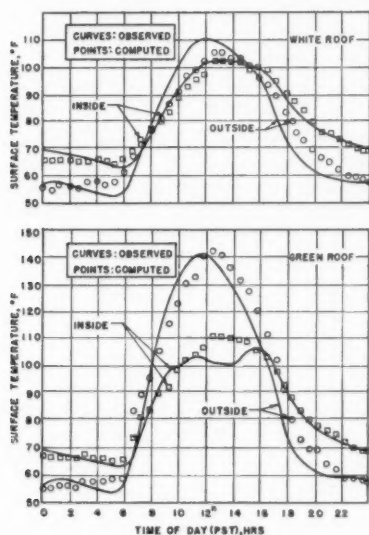


FIG. 7. DIURNAL VARIATION OF ROOF SURFACE TEMPERATURES (COMPUTED POINTS ARE ANALOGUE PREDICTIONS; CURVES ARE DIRECT MEASUREMENTS)

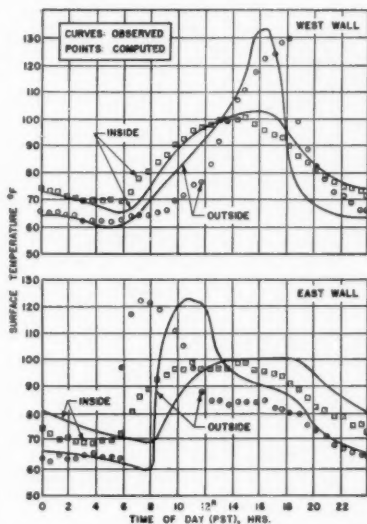


FIG. 8. DIURNAL VARIATION OF WEST AND EAST WALL SURFACE TEMPERATURES (COMPUTED POINTS ARE ANALOGUE PREDICTIONS; CURVES ARE DIRECT MEASUREMENTS)

various surfaces are given in Figs. 5 and 6 to aid in the interpretation of the analogue results.

Prediction of the sensible air conditioning load, Fig. 11, required to maintain the space at a constant temperature of 76 F was obtained as a second solution of the thermal circuit. The diurnal variations in heat flux at the inside surface of the various walls, the roof sections, and the floor are also given in Fig. 11.

*Boundary Inputs:* The difference between the computed and observed inputs, shown in Figs. 5 and 6, is due mainly to the approximation that the irradiation of a surface normal to the sun's rays is constant during the day and equal to the integrated mean value. The sharp rise in solar energy absorbed by the east wall after sunrise and the sharp drop at the west wall at sunset are due to relay limiters in the solar radiation input circuit.

A compromise (see Appendix A) in the choice of long-wave radiation potential resulted in considerably lower computer values of net loss during the day compared to the observed data shown in Fig. 6. This was compensated for partially by the low mid-day computer values of solar input.

*Prediction of Surface Temperatures:* The predicted roof and ceiling temperatures, shown in Fig. 7, are in good agreement with the measured values.

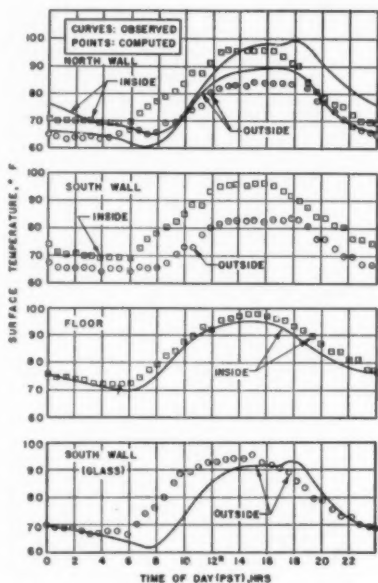


FIG. 9. DIURNAL VARIATION OF NORTH WALL, SOUTH WALL, FLOOR, AND GLASS SURFACE TEMPERATURES (COMPUTED POINTS ARE ANALOGUE PREDICTIONS; CURVES ARE DIRECT MEASUREMENTS)

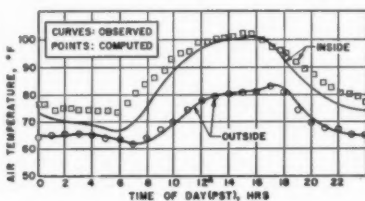


FIG. 10. DIURNAL VARIATION OF OUTSIDE AND INSIDE AIR TEMPERATURE (COMPUTED POINTS ARE ANALOGUE PREDICTIONS; CURVES ARE DIRECT MEASUREMENTS)

As indicated by Fig. 8, the predicted inside surface temperatures of the west wall agree well with observed temperatures. The predicted east wall temperature cycle agrees well in amplitude but is considerably out of phase, leading the observed temperature cycle by approximately 3 hr. The difference in phase for the outside surface temperature can be explained by the shading of the east wall of the test house for approximately  $2\frac{1}{2}$  hr after sunrise and by the sharp rise in the com-

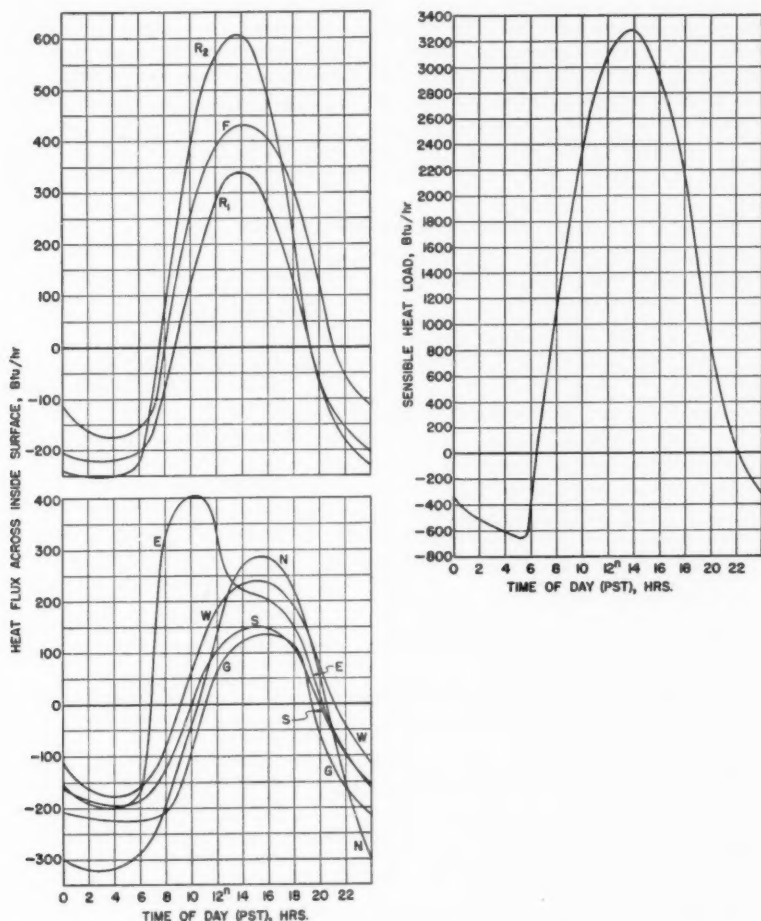


FIG. 11. DIURNAL VARIATION OF SENSIBLE AIR CONDITIONING LOAD TO MAINTAIN CONSTANT AIR TEMPERATURE (76°F) AND THE SINGLE SURFACE HEAT FLUXES (ANALOGUE PREDICTION) (HEAT FLUX "IN" IS POSITIVE)

puted radiation input immediately after sunrise. The inside surface temperature follows the inside air temperature rather closely and also indicates some response to the rising outside surface temperature during the morning.

The predicted north wall outside surface temperature cycle, shown in Fig. 9, compares favorably with the observed values except that they are as much as 5 F less during the afternoon. This difference may be accounted for by the fact that reflection from the north parapet was neglected. The predicted inside surface temperature cycle leads the observed cycle but is similar in amplitude. It may be noted that the predicted surface temperature cycle is approximately in phase with the inside air temperature cycle. No comparison can be made with the predicted south wall temperatures since observed data were not recorded. It can be seen, however, that in this case again the inside surface temperatures follow closely the inside air temperature pattern.

The predicted inside floor surface temperature cycle, shown in Fig. 9, agrees well with the observed data both in phase and amplitude. The predicted south

### NOMENCLATURE AND SYMBOLS

$A$ = heat transfer area, square feet.	<i>Subscripts</i>
$C$ = total heat capacity, Btu per Fahrenheit degree.	$a$ = air space.
$c_p$ = unit heat capacity (specific heat) at constant pressure, Btu per (pound) (Fahrenheit degree).	$b$ = roofing material.
$E$ = constant electrical potential, volts.	$bb$ = black body.
$e$ = time variable electrical potential, volts.	$c$ = convection.
$H$ = irradiancy, or incident radiant flux per unit area, Btu per (hour) (square foot).	$D$ = direct solar radiation.
$h$ = altitude angle, degrees.	$d$ = diffuse or scattered solar radiation.
$I$ = constant electric current, amperes.	$E$ = east wall.
$i$ = time variable electric current, amperes.	$e$ = electrical circuit element.
$Q$ = thermal energy, Btu.	$F$ = floor.
$q$ = heat flux, Btu per hour.	$g$ = ground.
$\bar{R}$ = radiosity, Btu per (hour) (square foot).	$G$ = glass.
$R'$ = linearized radiosity.	$h$ = overhang.
$t$ = surface temperature, Fahrenheit.	$i$ = inside.
$T$ = surface temperature, Fahrenheit absolute.	$I$ = insulated wall space.
$X$ = angle of incidence of the sun with respect to a surface, degrees.	$M$ = mean value.
$Z$ = azimuth angle, degrees.	$N$ = surface normal to sun's rays.
<i>Greek Letters</i>	$N$ = north wall.
$\alpha$ = absorptivity, dimensionless.	$n$ = normal drawn to a surface.
$\epsilon$ = emissivity, dimensionless.	$n$ = any surface.
$\theta$ = time	$o$ = outside.
$\sigma$ = Stefan-Boltzmann constant, $0.173 \times 10^{-8}$ , Btu per (hour) (square foot) (degree Fahrenheit absolute)*.	$o$ = horizontal surface.
$\tau$ = air temperature, Fahrenheit.	$p$ = pyrheliometer measurements.
	$p$ = interior plywood.
	$P$ = parapet.
	$r$ = radiation.
	$R_1$ = white roof section.
	$R_2$ = green roof section.
	$S$ = south wall.
	sky = atmospheric mass.
	$s$ = sun or solar.
	$s$ = sheathing.
	$t$ = thermal circuit element.
	$W$ = west wall.
	<i>Abbreviations</i>
	lw = long-wave radiation (also LW).
	sw = short-wave radiation.

wall glass temperatures run considerably higher than the measured values during the morning hours approaching the inside air temperature between 9:00 and 10:00 a.m. On the other hand, the measured values approach the outside air temperature. The peculiar increase in glass temperature observed beginning about 5:00 p.m. was probably caused by direct rays of the sun coming in under the overhang from the west shortly before sunset.

#### DISCUSSION AND CONCLUSIONS

Examination of the test house thermal circuit and results of the first analogue solution (uncontrolled inside air temperature) reveals the potency of the recording equipment heat source in establishing the space temperature. This heat source was estimated at 1280 Btu/hr which would be sufficient to cause an air temperature rise rate of over 50 F per hr if no heat losses occurred. The major heat loss from the space during the day was through the floor, north wall, and south wall glass area. At night the greatest losses were due to heat transfer through the roof and the glass area. As one would expect, the insulated west wall was responsible for the least total heat loss from the space and the green roof section was responsible for the greatest heat gain with the exception of the heat source in the house. The heat source due to diffuse radiation was not accounted for in the circuit at this time. It may be included by connecting the output of current generators to each of the inside surface potentials. It was estimated that the floor and north wall would absorb diffuse solar radiation at an approximate rate of 250 Btu/hr and the remaining surfaces together approximately 120 Btu/hr during the day. Additional solutions are planned for the future to determine the importance of this source.

There appears to be a definite discrepancy in the predicted glass temperature (Fig. 9). Indications are that the outside glass surface resistances were much too large. Increased heat loss through the glass during the day (a more reasonable result) would have reduced the daytime inside air temperature prediction.

In general, it may be stated that the predictions of the thermal behavior of the test house are good. All of the previous discussion simply points to the fact that the thermal circuit solution can only be as good as the representation of the actual physical system and the ability to evaluate quantitatively the circuit parameters and boundary conditions. The value of the thermal circuit technique of analysis lies in the ability to see the total picture at once and to estimate the importance of the separate influences. The electric analogue solution has demonstrated a high degree of flexibility and economy of effort in the rapid quantitative determination of the influence of many variables on the behavior of a complex thermal system.

For the first solution, actual heat fluxes across the various surface boundaries were not computed from the analogue data because the method of calculation could result in large errors. The heat flux across any inside surface boundary is determined by taking the difference (a small number) of two relatively large quantities (surface and air potentials) measured above ground and dividing by the convective resistance. It can be shown that errors of as little as 1 percent in each of the temperatures involved in the computation of heat flux can result in errors of 100 percent or more for different surfaces and times involved in these studies. For the second solution, involving a constant inside air potential, the inside surface potentials were measured with respect to this constant value resulting in a much more accurate determination of heat fluxes. In addition, the measuring unit was modi-

fied to allow the expansion of the voltage range required over the full scale of the oscillograph.

It is estimated that lumping errors, instrumentation errors, leakage errors, all put together can result in errors of the order of 5 percent in any one quantity computed. The greatest area of interpretation lies in the original postulates and the evaluation of circuit parameters.

The studies described are the first phase in the application of electric analogue techniques to the determination of space heating and cooling requirements. Additional studies currently being conducted and others planned for the future involve: (1) reduction of circuit elements (simplification of lumping); (2) determination of the importance of separate influences such as radiation inputs at the various inside surfaces, ventilation or heat sinks, inputs at the exterior surfaces due to solar and longwave radiation; (3) determination of the effectiveness of shading devices, high solar reflectivity paints, capacitance and/or insulation in different locations, and evaporative cooling of roofs in reducing the actual cooling load; (4) study of methods for including the influence of landscaping in the thermal circuit; (5) study of the effect of the physical variables on an occupant by introducing into the thermal circuit the network representing the human thermal system; and (6) the introduction of equipment characteristics and control programming to the network.

#### ACKNOWLEDGMENTS

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## APPENDIX A

### Representation of Boundary Conditions

An energy balance at any exterior surface of the test house may be expressed by

$$Ak \left( \frac{\partial t}{\partial x} \right)_{x=0} = q_s + \frac{t_{1w} - t_o}{R_r} + \frac{\tau_o - t_o}{R_o} \quad \text{. . . . . (A-1)}$$

#### 1. SOLAR RADIATION INPUT

The first term of the right hand member of Equation A-1 represents the net radiation exchange with the sun. The equivalent black body temperature of the sun is approximately 10,000 F, therefore, the net radiation exchange between the sun and wall surface is independent of the wall temperature for all practical purposes, and may be represented as a heat input  $q_s$ .

The solar heat input consists of two parts: that due to direct solar irradiation and that due to diffuse solar irradiation. In its transmission through the earth's atmosphere a fraction of the radiation is scattered by particles and air molecules and reflected back toward earth as diffused radiation, a fraction is absorbed mainly by water vapor, carbon dioxide, and ozone and the balance passes through as direct radiation. The reduction of direct solar radiation is a function of the length of path and conditions through the atmosphere. Therefore, solar irradiation of a surface on earth, normal to the sun's rays would be expected to vary with the altitude of the sun at the particular locality and with the nature of the atmospheric mass in the path of the sun's rays such as quantity and distribution of water vapor, clouds, dust particles, carbon dioxide, and other chemical constituents. Due to strong absorption by  $\text{CO}_2$  and  $\text{H}_2\text{O}$  beyond  $2.3 \mu$  and by  $\text{O}_3$  at  $0.29 \mu$  approximately 95 percent of the solar radiant power received at the earth's surface is distributed between  $0.3$  and  $2.5 \mu$ .

The total solar irradiation of a surface on earth oriented in any fashion may be represented by

$$(H_T)_n = (H_D)_n \cos X_n + F_{n,sky} H_d \quad \text{. . . . . (A-2)}$$

when it is postulated that diffuse solar radiation is of uniform intensity over the hemisphere, and that solar radiation received due to reflection from the ground or other surfaces is negligible.

The direct solar irradiation of a surface oriented in any fashion (the first term of the right hand member of Equation A-2) may be evaluated as follows:

$$\text{By definition,} \quad (H_D)_o = H_p - H_d \quad \text{. . . . . (A-3)}$$

Let,

$$\frac{H_d}{H_p} = \beta$$

then,

$$(H_D)_o = H_p(1 - \beta) \quad \text{. . . . . (A-4)}$$

also,

$$(H_D)_o = (H_D)_n \cos X_o \quad \text{. . . . . (A-5)}$$

where,

$$X_o = 90 - h_s \quad \text{. . . . . (A-6)}$$

Combining Equations A-4, A-5, and A-6 and solving for  $(H_D)_N$  results in

$$(H_D)_N = \frac{(H_D)_0}{\cos(90 - h_s)} = \frac{H_p(1 - \beta)}{\sin h_s} \dots \dots \dots (A-7)$$

therefore,

$$(H_D)_N = \frac{H_p(1 - \beta)}{\sin h_s} \cos X_n \dots \dots \dots (A-8)$$

The angle of incidence  $X$  may be found for any oriented surface any place on earth in terms of the sun's declination and hour angle, and local apparent time<sup>1</sup>.

$$\cos X_n = \sin \delta_s \sin \delta_n + \cos \delta_s \cos \delta_n \cos(\theta_n - 15\theta) \dots \dots (A-9)$$

where,  $15\theta$  is the local hour angle of the sun,  $\theta$  being given as hours after midnight. From the astronomical triangle,

$$\sin \delta_n = \sin \phi \sin h_n + \cos \phi \cos h_n \cos Z_n \dots \dots \dots (A-10)$$

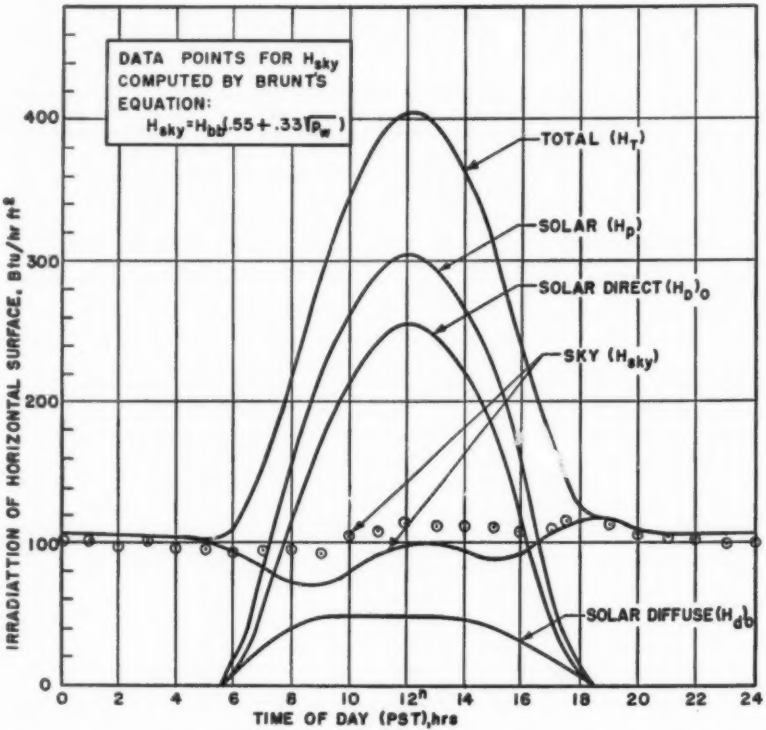


FIG. A-1. OBSERVED IRRADIATION OF A HORIZONTAL SURFACE FOR THE 24-HR PERIOD BEGINNING SEPTEMBER 8, 1953

and,

$$\sin \theta_n = \frac{\cos h_n \sin Z_n}{\cos \delta_n} \quad (A-11)$$

Substituting for  $\cos X$  in Equation A-8 results in

$$(H_D)_n = \frac{H_p(1-\beta)}{\sin h_s} [\sin \delta_s \sin \delta_n + \cos \delta_s \sin \delta_s \cos(\theta_n - 15\theta)] \quad (A-12)$$

For purposes of the electric analogue solution, Equation A-12 may be simplified by making the following approximations:

$$\frac{H_p}{\sin h_s} \cong \text{constant for a 1 day cycle}$$

$$\frac{H_p}{\sin h_s} \cong \frac{\int_{\theta_{\text{sur}}}^{\theta_{\text{sun}}} \frac{H_p}{\sin h_s} d\theta}{\theta_{\text{sun}} - \theta_{\text{sur}}} = H_M \cong H_p \text{ (at noon)}$$

$$\beta \cong \text{constant for a 1 day cycle.}$$

$$\delta_s \cong \text{constant for a 1 day cycle.}$$

$$\delta_n \cong (\delta_n)_M \text{ or mean value for the particular day of interest.}$$

where,

sun = sunset.

sur = sunrise.

On the basis of these approximations Equation A-12 may now be written

$$(H_D)_n = A + B \cos(\theta_n - 15\theta) \quad (A-13)$$

where,

$$A = \frac{H_p(1-\beta)}{\sin h_s} (\sin \delta_s \sin \delta_n) \quad (A-14)$$

$$B = \frac{H_p(1-\beta)}{\sin h_s} (\cos \delta_s \cos \delta_n) \quad (A-15)$$

For a horizontal surface Equation A-13 reduces to

$$(H_D)_o = \frac{H_p}{\sin h_s} (1-\beta) \sin h_s \quad (A-16)$$

$\sin h_s$  may be represented as a cosine function as follows:

$$\sin h_s = G - F \cos 15\theta \quad (A-17)$$

where

$$G = \frac{[\sin(h_s)_{12}] \cos 15\theta_{\text{sur}}}{1 + \cos 15\theta_{\text{sur}}} \quad (A-18)$$

$$F = \frac{\sin(h_s)_{12}^n}{1 + \cos 15\theta_{\text{sur}}} \quad (A-19)$$

Equations A-13 and A-16 must be limited to only the time of day when the sun sees the surface. A function  $\psi(\theta)$  was used as a multiplier to modify these equations, accounting for sunrise and sunset by making

$$\psi(\theta) = 1 \text{ for } 5.6 < \theta < 18.4 \text{ hr}$$

$$\psi(\theta) = 0 \text{ for } 5.6 > \theta > 18.4 \text{ hr}$$

The function  $\psi(\theta)$  may also be used to account for fixed obstructions and/or shading devices. In addition, the equations must be limited to positive values.

Equations A-13 and A-16 were evaluated using observed values of irradiation of a horizontal surface for the 24-hr period beginning September 8, 1953. Fig. A-1 presents

the total irradiation of a horizontal surface and the separate contributions due to the solar and low temperature sources. The irradiation due to the sky (atmospheric mass) was obtained by taking the difference between the flat plate total hemispherical radiometer and the Eppley pyrheliometer. Also shown are points computed by Brunt's equation which expresses  $H_{sky}$  for a clear atmosphere as an empirical function of black body radiation at air temperature and water vapor partial pressure at ground level.

The results of the calculations follow:

$$\begin{aligned}\frac{H_p}{\sin h_s} &= (H_p)_M = 302 \text{ Btu per (hr) (sq ft)} \\ \beta &= 0.18 \\ \delta_s &= (\delta_s)_M = 5^\circ 42' \text{ (Gurley Ephemeris, 1953 Edition)} \\ \theta_{sur} &= 5.60 \text{ hr} \\ \theta_{sun} &= 18.4 \text{ hr} \\ (h_s)_{12}^N &= 62.0 \text{ deg}\end{aligned}$$

$$\text{therefore,} \quad G = \frac{(0.883) (0.1045)}{1 + 0.1045} = 0.0836$$

$$F = \frac{0.883}{1 + 0.1045} = 0.799$$

$$\text{then,} \quad (H_D)_o = (302) (0.82) [0.0836 - 0.799 \cos 15\theta]$$

$$\text{therefore,} \quad (H_D)_o = 20.7 - 198 \cos 15\theta$$

A summary of the calculation of Equation A-13 for the various exposed vertical surfaces is given in Table A-1.

TABLE A-1. CALCULATIONS OF DIRECT SOLAR IRRADIATION OF EXPOSED VERTICAL SURFACES

EXPOSED SURFACE	$\frac{A}{\text{BTU}/(\text{HR})}$ (SQ FT)	$\frac{B}{\text{BTU}/(\text{HR})}$ (SQ FT)	$\theta_n$ DEG	DIRECT SOLAR IRRADIATION ( $H_D$ ), BTU/(HR) (SQ FT)
West Wall.....	0	246	270	$-246 \sin 15\theta$
East Wall.....	0	246	90	$246 \sin 15\theta$
North Wall.....	17.88	138	0	$17.88 + 138 \cos 15\theta$
South Wall.....				zero, due to eave overhang

The diffuse solar irradiation of a surface oriented in any fashion (the second term of the right hand member of Equation A-2) may be evaluated as follows:

$$\text{by definition} \quad (H_d)_o = H_p \beta, \quad \text{but} \quad \frac{H_p}{\sin h_s} \cong (H_p)_M$$

$$\text{therefore,} \quad (H_d)_o = (H_p)_M \beta \sin h_s \quad \dots \quad (\text{A-20})$$

Combining Equations A-20 and A-17

$$(H_d)_o = (H_p)_M \beta [G - F \cos 15\theta] \quad \dots \quad (\text{A-21})$$

Equation A-21 was applied to the vertical surfaces using a shape modulus of 0.5 for the unobstructed east and west surfaces with respect to the sky, 0.3 for the north wall due to partial obstruction by the parapet, and 0.3 for the south wall due to partial obstruction by the eave overhang. A summary of the calculation of Equation A-21 for the various exposed exterior surfaces is given as follows:

$$\begin{aligned}
 \text{Roof:} & \quad (H_a)_o = 4.55 - 43.4 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 \text{West Wall:} & \quad (H_a)_w = 2.28 - 21.7 \cos 15\theta \\
 \text{East Wall:} & \quad (H_a)_e = 2.28 - 21.7 \cos 15\theta \\
 \text{North Wall:} & \quad (H_a)_n = 1.36 - 13.0 \cos 15\theta \\
 \text{South Wall:} & \quad (H_a)_s = 1.36 - 13.0 \cos 15\theta
 \end{aligned}$$

The solar heat inputs to each of the exposed exterior surfaces were finally established by summing the direct and diffuse irradiation, and reducing the sum in accordance with the fraction of incident radiation absorbed. One value of absorptivity representing an overall value covering the solar spectrum was used for both the direct and diffuse radiation components. The estimated solar radiation properties of the surfaces involved in this study are given in Table A-2.

TABLE A-2. OVERALL SHORT-WAVE RADIATION PROPERTIES OF EXPOSED SURFACES

SURFACE	EMISSIONITY OR ABSORPTIVITY	TRANSMISSIVITY (DIFFUSE)
Exterior, buff color painted.....	0.48	(Opaque surface)
Green Rolled Roofing.....	0.88	(Opaque surface)
White Painted Roof Section Partially Discolored.....	0.40	(Opaque surface)
Standard $\frac{1}{8}$ in. Window Glass.....	0.06	0.79

The solar heat input equations in thermal units are then converted to current inputs, the analogous electrical quantity, by multiplying each equation through by the scale factor,  $\frac{1}{16 \times 10^6}$ . The voltage input to the current generators which will result in the correct current outputs are determined by

$$e_{in} = i_{out} R_x \quad (A-22)$$

where,

$R_x$  = the external feedback resistor of the current generator.

A summary of the solar heat and current input equations and the current generator voltage inputs follows. Note that the voltage inputs are all negative values because the current generator input and output are 180 deg out of phase.

#### West Wall

$$\begin{aligned}
 (q_s/A)_w &= 1.094 - 10.42 \cos 15\theta - \psi(\theta) 118 \sin 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_w &= 0.00581 - 0.0554 \cos 15\theta - \psi(\theta) 0.627 \sin 15\theta, \text{ ma (milliamperes)} \\
 &\quad \text{for } R_x = 146,000 \text{ ohms} \\
 (e_{in})_w &= 0.848 - 8.09 \cos 15\theta - \psi(\theta) 91.5 \sin 15\theta, \text{ volts}
 \end{aligned}$$

#### East Wall

$$\begin{aligned}
 (q_s/A)_e &= 1.094 - 10.42 \cos 15\theta + \psi(\theta) 118 \sin 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_e &= 0.00581 - 0.0554 \cos 15\theta + \psi(\theta) 0.627 \sin 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 146,000 \text{ ohms} \\
 (e_{in})_e &= 0.848 - 8.09 \cos 15\theta + \psi(\theta) 91.5 \sin 15\theta, \text{ volts}
 \end{aligned}$$

#### North Wall

$$\begin{aligned}
 (q_s/A)_n &= [8.58 + 66.2 \cos 15\theta] \psi(\theta) + 0.652 - 6.24 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_n &= [0.0676 + 0.521 \cos 15\theta] \psi(\theta) + 0.00513 - 0.0491 \cos 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 167,000 \text{ ohms} \\
 (e_{in})_n &= [11.29 + 87.0 \cos 15\theta] \psi(\theta) + 0.856 - 8.20 \cos 15\theta, \text{ volts}
 \end{aligned}$$

*White Roof*

$$\begin{aligned}
 (q_s/A)_{R1} &= 10.1 - 96.5 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_{R1} &= 0.0458 - 0.437 \cos 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 200,000 \text{ ohms} \\
 (e_{in})_{R1} &= 9.16 - 87.4 \cos 15\theta, \text{ volts}
 \end{aligned}$$

*Green Roof*

$$\begin{aligned}
 (q_s/A)_{R2} &= 22.2 - 212 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_{R2} &= 0.1221 - 1.166 \cos 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 75,000 \text{ ohms} \\
 (e_{in})_{R2} &= 9.16 - 87.4 \cos 15\theta, \text{ volts}
 \end{aligned}$$

*South Wall*

$$\begin{aligned}
 (q_s/A)_B &= 0.652 - 6.24 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_B &= 0.00278 - 0.0266 \cos 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 188,000 \text{ ohms} \\
 (e_{in})_B &= 0.523 - 5.00 \cos 15\theta, \text{ volts}
 \end{aligned}$$

*Window Glass (South Wall)*

$$\begin{aligned}
 (q_s/A)_G &= 0.0816 - 0.780 \cos 15\theta, \text{ Btu per (hr) (sq ft)} \\
 (i_s)_G &= 0.000295 - 0.00282 \cos 15\theta, \text{ ma} \\
 &\quad \text{for } R_x = 1.75 \text{ meg ohms} \\
 (e_{in})_G &= 0.523 - 5.00 \cos 15\theta, \text{ volts}
 \end{aligned}$$

A solar input circuit shown in Fig. A-2 was devised to solve the solar heat input equations delivering a time-variable current proportional by the proper scale factor to the incident solar radiation absorbed by each surface.

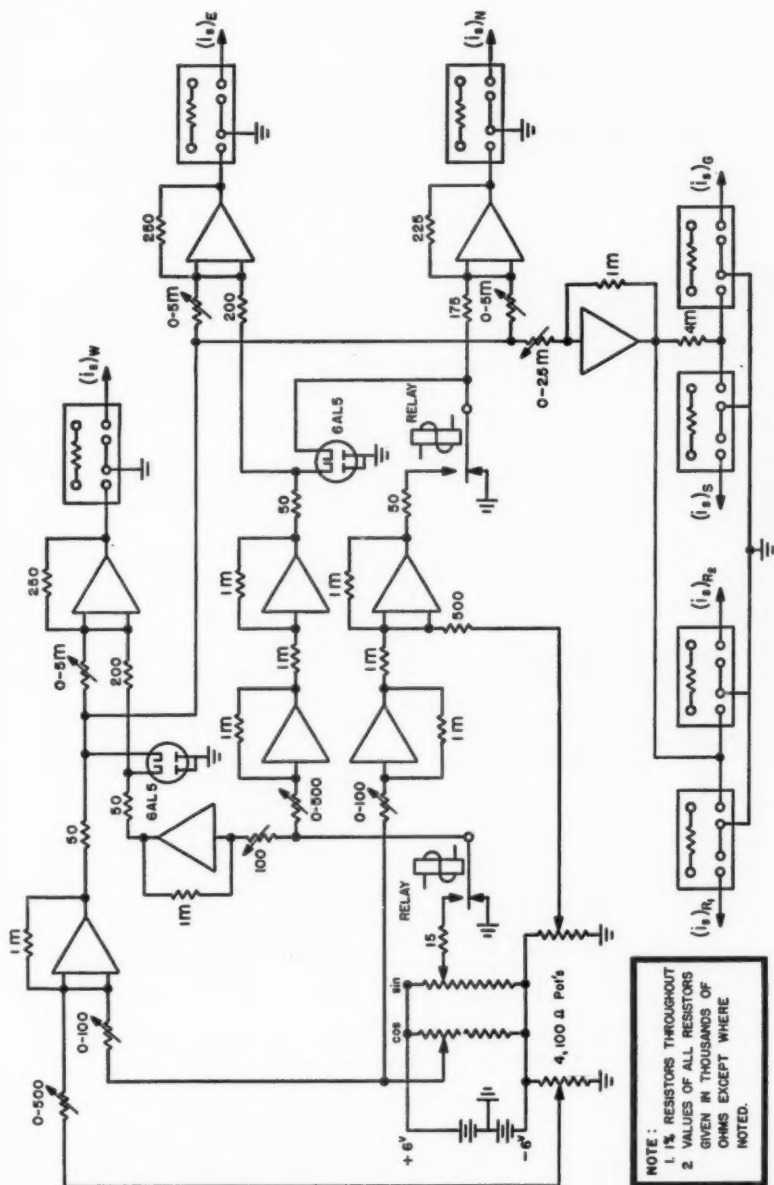
The current generators are connected to the network through a switching circuit, shown in Fig. A-3. Through the switching circuit, each current generator input terminal may be connected to either ground or an input potential. When the current generator inputs are grounded the output of any current generator, selected through a switch, is connected to ground through a microammeter, the other generator outputs being connected directly to ground. Thus, the current generators may be rapidly checked for balance before a solution is obtained. Switching to the operate position connects all generator input terminals to the input potential signals and the output terminals to the proper point in the R-C network.

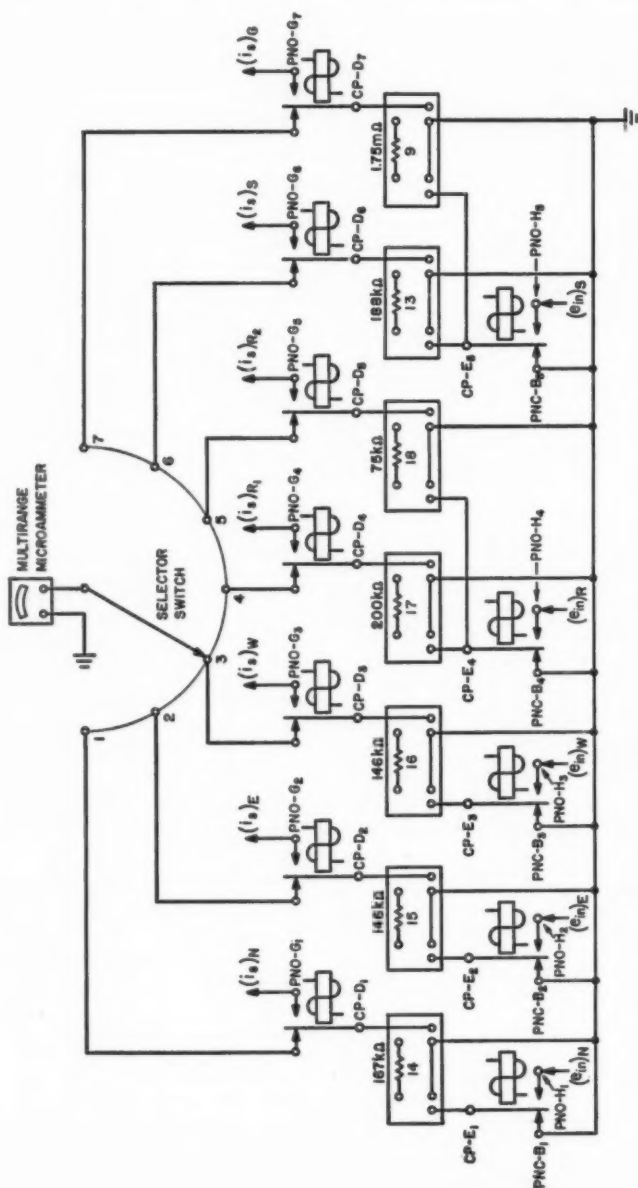
The input functions were measured by means of d-c amplifiers and recording oscillographs. Fig. A-4 presents reproductions of the solar inputs recorded on tapes.

## 2. LOW TEMPERATURE, LONG-WAVE RADIATION EXCHANGE AT OUTSIDE SURFACES

The second term of the right hand member of Equation A-1 represents the net long-wave radiation exchange between an exposed exterior surface and the surroundings. This involves radiation exchange with the atmospheric mass, the ground, and other objects that see the surface. The roof of the test house exchanged long-wave radiation with the atmospheric mass only; the east and west walls with the ground and atmosphere; the north wall with the ground, parapet, and atmosphere; and the south wall with the ground, eave overhang, and atmosphere.

The only important thermal emitters in the clear atmospheric mass are water vapor and carbon-dioxide. As an approximation, these gases may be thought of as being emitters or absorbers over spectral bands of finite width. Bands of importance for water vapor exist at wave lengths of approximately  $2.6 \mu$ ,  $6.3 \mu$ ; and at a series of wave-lengths greater than  $18 \mu$ . Carbon-dioxide has important emission bands at approximately  $4.3 \mu$  and  $15 \mu$ . The peak emission of the other radiators in the surroundings such as the ground and buildings occurs at approximately  $9.5 \mu$ . For the purposes of





**Notes:** (1) CP refers to center pole of relay; PNC refers to pole normally closed; PNO refers to pole normally open. (2) *Operate* position switch (not shown) connects PNC with PNO. (3) *Calibrate* position switch (not shown) connects CP point selected through meter to ground. All other CP points are connected to corresponding PNC points which are grounded internally. (4) dash letters at connection points are for identification only.

FIG. A-3. CURRENT GENERATOR SWITCHING CIRCUIT

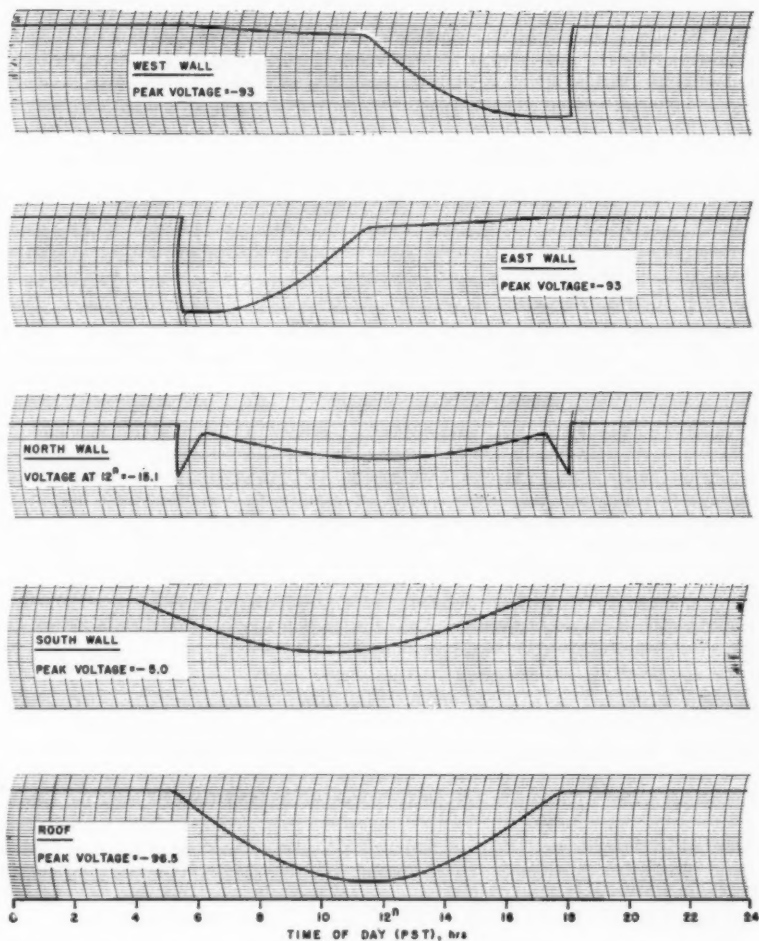


FIG. A-4. OSCILLOGRAMS OF SOLAR RADIATION INPUT FUNCTIONS; MEASUREMENTS WERE MADE AT THE CURRENT GENERATOR INPUTS

this investigation the long-wave radiation exchange resistances are based on *overall* long-wave radiation properties given in Table A-3.

It is apparent that a single long-wave radiation potential of the surroundings for all of the exposed surfaces does not exist. Separate surroundings potentials might be defined for each surface where the surroundings are similar. This would have required a potential for the roof sections corresponding to an equivalent sky temperature; a potential for the east and west walls corresponding to some mean temperature of the sky and ground; a potential for the north wall corresponding to some mean temperature

TABLE A-3. OVERALL LONG-WAVE RADIATION PROPERTIES OF EXTERIOR SURFACES

SURFACE	LONG WAVE EMISSIVITY OR ABSORPTIVITY
Exterior Buff Color Painted.....	0.85
Green Rolled Roofing.....	0.90
White Painted Roof Section.....	0.85
Parapet (Painted Stucco).....	0.85
Ground (Roof of Engineering Building).....	0.90
Standard $\frac{1}{8}$ in. Window Glass.....	0.85

of the sky, ground and parapet; and a potential for the south wall corresponding to some mean temperature of the sky, ground, and eave overhang. Each of these potentials would be time-variable quantities. To simplify this complex situation a single time-variable potential corresponding to a specified fraction of the ambient air temperature was used as a first approximation and the radiation resistance for each surface was calculated accordingly. It was not feasible to use the air temperature as the potential because that would have resulted in a net energy transfer to the surfaces at night when in reality there was always a net loss to the surroundings. Without substantial justification, it was decided to assume the long-wave potential ( $t_{lw}$ ) as being equal to a constant fraction of the air temperature where the fraction was equal to the ratio of equivalent sky temperature to air temperature at 12 noon. Thus,

$$t_{lw} = \frac{(t_a)_{12^n}}{(\tau_a)_{12^n}} \tau_a \dots \dots \dots (A-23)$$

Each of the long-wave radiation resistances for any surface  $n$  were defined as follows:

$$\frac{t_{lw} - (t_a)_n}{(R_{ra})_n} = \text{Net long-wave exchange for surface } n \dots \dots (A-24)$$

The radiation resistance,  $(R_{ra})_n$ , determined for each exposed surface represented a mean value based on calculations for 12 noon and 12 midnight. A summary of the net-exchange equations is given as follows:

*West Wall*

$$(q_r)_W = A_W [F_{Wsky}(\alpha_{lw})_W H_{sky} + F_{Wg}(\epsilon_{lw})_g(\alpha_{lw})_W \sigma T_g^4 - (\epsilon_{lw})_W \sigma T_W^4]$$

*East Wall*

$$(q_r)_E = A_E [F_{Esky}(\alpha_{lw})_E H_{sky} + F_{Eg}(\epsilon_{lw})_g(\alpha_{lw})_E \sigma T_g^4 - (\epsilon_{lw})_E \sigma T_E^4]$$

*North Wall*

$$(q_r)_N = A_N [F_{Nsky}(\alpha_{lw})_N H_{sky} + F_{Ng}(\epsilon_{lw})_g(\alpha_{lw})_N \sigma T_g^4 + F_{NP}(\epsilon_{lw})_P(\alpha_{lw})_N \sigma T_P^4 - (\alpha_{lw})_N \sigma T_N^4]$$

## South Wall

$$(q_r)_S = A_S[F_{Ssky}(\alpha_{1w})_SH_{sky} + F_{Sg}(\epsilon_{1w})_g(\alpha_{1w})_S\sigma T_g^4 + F_{Sh}(\epsilon_{1w})_h(\alpha_{1w})_S\sigma T_h^4 - (\alpha_{1w})_S\sigma T_N^4]$$

## Window Glass (South Wall)

$$(q_r)_G = A_G[F_{Gsky}(\alpha_{1w})_GH_{sky} + F_{Gg}(\epsilon_{1w})_g(\alpha_{1w})_G\sigma T_g^4 + F_{Gh}(\epsilon_{1w})_h(\alpha_{1w})_G\sigma T_h^4 - (\alpha_{1w})_G\sigma T_G^4]$$

## Roof

$$\text{White:} \quad (q_r)_{R1} = A_{R1}[(\alpha_{1w})_{R1}H_{sky} - (\epsilon_{1w})_{R1}\sigma T_{R1}^4]$$

$$\text{Green:} \quad (q_r)_{R2} = A_{R2}[(\alpha_{1w})_{R2}H_{sky} - (\epsilon_{1w})_{R2}\sigma T_{R2}^4]$$

where

$$\begin{aligned} F_{Wsky} \text{ and } F_{Wsky} &= 0.50 \\ F_{Ng}, F_{Nsky}, F_{Ssky}, \text{ and } F_{Ssky} &= 0.30 \\ F_{Wg}, F_{Wg}, F_{Sg}, \text{ and } F_{Sg} &= 0.50 \\ F_{NP} &= 0.40 \text{ and } F_{Sh} = 0.20 \end{aligned}$$

A summary of the long-wave radiation resistances is given in Table A-4. Glass was considered to be opaque to long-wave radiation.

## 3. CONVECTIVE EXCHANGE AT OUTSIDE SURFACES

The third term of the right hand member of Equation A-1 represents the convective exchange between the exposed surfaces and the air mass. The air potential ( $\tau_o$ ) was assumed to be the same for all surfaces and equal to the air temperature actually measured during the test period. The convective resistances were evaluated by using the simplified forced and free convection equations for air flow past flat plates summarized in THE GUIDE. The values of conductance obtained from these equations were adjusted to account for the roughness of the real surfaces. For a given surface in air, the convective resistance is influenced mainly by the air velocity over the surface making it variable with time.

During September, at the test house location, the wind generally blows inland from the ocean or westerly at approximately 1 to 10 mph reaching a peak in mid-afternoon (3:00 to 4:00 p.m.). For all hours before about 9:00 a.m. and after 8:00 p.m., the air velocity remains low—of the order of 1 mph and the direction during the night is toward the ocean or from the east. Assuming an air velocity of approximately 500 fpm (5 to 6 mph) which was about the average measured on September 8, 1953, an average conductance spacewise, based on parallel turbulent flow over flat rough surfaces is approximately 2.0 Btu per (hr) (sq ft) (F deg). During the night when the convective losses were substantially by free convection and laminar flow, a value of 0.70 Btu per (hr) (sq ft) (F deg) was estimated for the vertical walls and 1.0 Btu per (hr) (sq ft) (Fahrenheit degree) for the roof. Because the house itself acts as an obstruction with respect to flow over the east wall during the day and the wind shifts from west to east during the night, an average value of conductance, 1.35 Btu per (hr) (sq ft) (F deg) was used for the east wall.

Convective resistance is defined as the driving potential required per unit heat flux, or,

$$(R_{Co})_n = \frac{\tau_o - (t_o)_n}{q_n} \quad \text{. . . . . (A-25)}$$

By Newton's Law of Cooling

$$q_n = h_c A_n [\tau_o - (t_o)_n] \quad \text{. . . . . (A-26)}$$

TABLE A-4. SUMMARY OF LONG-WAVE RADIATION AND CONVECTION RESISTANCES

SURFACE	OUTSIDE LONG-WAVE RADIATION RESISTANCES, ( $R_{ro}$ )		OUTSIDE CONVECTIVE RESISTANCES, ( $R_{co}$ )				INSIDE CONVECTIVE RESISTANCES, ( $R_{ci}$ )	
			DAY		NIGHT			
	F DEG (BTU/HR)	1000 OHMS	F DEG (BTU/HR)	1000 OHMS	F DEG (BTU/HR)	1000 OHMS	F DEG (BTU/HR)	1000 OHMS
West Wall.....	0.0700	1120	0.00589	94.1	0.00168	269	0.0236	378
East Wall.....	0.0700	1120	0.00871	140.	0.00871	140	0.0236	378
North Wall.....	0.0587	940	0.00397	63.5	0.01132	181	0.01588	254
South Windows.....	0.125	2000	0.00731	117	0.0210	336	0.0293	469
White Roof Section.....	0.188	3000	0.01022	164	0.0248	397	0.0348	555
Green Roof Section.....	0.0262	420	0.00690	110	0.0138	221	0.0460	735
Floor.....	0.0201	323	0.00569	91.0	0.0137	182	0.0379	606
			0.0208	333	0.0208	333	0.0104	166

$$(R_{co})_n = \frac{1}{h_o A_n} \dots \dots \dots (A-27)$$

A summary of the outside convective resistances for the various surfaces is given in Table A-4.

#### 4. CONVECTIVE EXCHANGE AT INSIDE SURFACES

The air movement within the enclosed space was of the order of 10 fpm or less. The conductances were estimated on the basis of free convection. For a particular location, free convection conductances are mainly a function of surface geometry and orientation within the air mass, and temperature difference between the surface and fluid. The variation in temperature differences for the inside surfaces and air mass are not large and furthermore  $h_{ei}$  varies approximately as  $(\Delta T)^{0.25}$ . Therefore, the inside surface conductances were considered to be constant for the 24 hr cycle and equal to the following values:

$$\begin{aligned} h_{ei} \text{ (for vertical surfaces)} &= 0.50 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).} \\ h_{ei} \text{ (for horizontal surfaces facing down)} &= 0.30 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).} \\ h_{ei} \text{ (for horizontal surfaces facing up)} &= 0.60 \text{ Btu per (hr) (sq ft) (Fahrenheit degree).} \end{aligned}$$

A summary of the inside surface convective resistances in thermal and electrical units computed in accordance with Equation A-27 is given in Table A-4.

#### 5. HEAT SOURCES AND SINKS

Heat sources and sinks representing heating and cooling devices, infiltration or leakage, transmitted radiation and electrical appliances can be readily incorporated in the thermal and electrical analogous circuits. In this investigation the recording instruments located in the test house were a significant heat source of approximately 375 watts or 1280 Btu per (hr). This required a constant current source of 0.080 milliamperes connected to the inside air capacitor. The required external feedback resistor required with a constant current generator input of 23.5 volts was found to be 294,000  $\Omega$  by Equation A-22.

Time variable heat sources such as the transmission of direct solar radiation can be included by connecting current generators to the irradiated surfaces and providing them with the appropriate inputs, similar to the treatment of solar inputs to exterior surfaces. Diffuse solar radiation transmitted by the south windows was neglected in this study. Infiltration was also considered negligible.

### APPENDIX B

#### Calculation of Circuit Parameters

The calculation of the conduction path resistances and capacitances will be summarized in this Appendix.

##### 1. THERMAL RESISTANCES

The Fourier law of heat conduction for unidirectional flow is given as

$$dq = -k dA \frac{dt}{dx} \dots \dots \dots (B-1)$$

where,

$dq$  = heat flux in x-direction.  
 $k$  = thermal conductivity.

$dA$  = elementary area perpendicular to heat flow.

$\frac{dt}{dx}$  = temperature gradient in the x-direction.

For the temperature gradient, thermal conductivity, and heat transfer area constant over a path length ( $l$ ) and the temperature uniform over an area ( $A$ ) at any point along the conduction path, Equation B-1 may be written as

$$q = \frac{kA}{l} (t_1 - t_2). \quad \text{. . . . . (B-2)}$$

By definition, thermal resistance is the driving potential (temperature) per unit heat flux,

$$R = \frac{t_1 - t_2}{q} = \frac{l}{kA}, \quad \frac{F \text{ deg}}{(\text{Btu/hr})}. \quad \text{. . . . . (B-3)}$$

In evaluating the conductive resistances of the test house circuit:

1. For each material involved an average value of  $k$  for the temperature range expected was used;
2. An *effective* value of  $k$  was used for the insulation material;
3. For the rolled roofing material where no definite value of thickness was known, values of conductance ( $c = \frac{k}{l}$ ), based on nominal thickness, were used; and,
4. Resistance through air space was determined on the basis of equivalent conductance.

The values of thermal conductivities and conductance for the materials involved in this study are given in Table B-1. A summary of the conduction path resistances for all sections of the circuit are given in Table B-2.

## 2. THERMAL CAPACITANCE

Thermal capacity, or the heat necessary to cause unit change in temperature of the mass involved may be defined as

$$C = \frac{Q}{t_f - t_i} = \frac{\int_{t_i}^{t_f} c_p \gamma V dt}{t_f - t_i} \quad \text{. . . . . (B-4)}$$

where,

$Q$  = thermal energy required to raise the temperature of a given mass from  $t_i$  to  $t_f$ .

If the unit heat capacity  $c_p$  and the density ( $\gamma$ ) are constant over the temperature interval considered or if appropriate mean values of these quantities are used, Equation B-4 may be written as

$$C = c_p \gamma V = c_p \gamma Al \quad \text{. . . . . (B-5)}$$

where

$l$  = the length and  $A$  is the area of a particular lump.

Values of unit heat capacity and density for the materials involved are given in Table B-1. A summary of the conduction path capacitors is given in Table B-3.

TABLE B-1. THERMAL PROPERTIES OF MATERIALS USED IN THIS INVESTIGATION

MATERIAL	THERMAL CONDUCTIVITY			UNIT HEAT CAPACITY			MASS DENSITY		
	k <sup>a</sup>	TEMP. RANGE F DEG	REF.	C <sub>p</sub> BTU/LB F DEG	TEMP. RANGE F DEG	REF.	LB/(CU FT)	TEMP. RANGE F DEG	REF.
		THERMAL CONDUCTANCE							
	b	TEMP. RANGE F DEG	REF.						
		6.5 1.2		ASHVE ASHVE				70	
Douglas Fir Wood.....	0.065	75	DFPA	0.5	149	Hawley	34	75	DFPA
Rockwool Insulation.....	0.025	90	ASHVE	0.201	271	Wilkes	10	90	ASHVE
Window Glass.....	0.416	68	I.C.T.	0.212		Corning	150		Corning
	0.441	212	I.C.T.	0.20					
Roofing (Asphalt Shingles).....	6.5		ASHVE				70		ASHVE
Vertical Air Space ¾.....	1.2		ASHVE						

<sup>a</sup> In Btu per (hr) (sq ft) (F deg per foot thick).<sup>b</sup> Figures in this column are in Btu per (hr) (sq ft) (Fahrenheit degree).

TABLE B-2—SUMMARY OF TEST HOUSE CONDUCTION PATH RESISTORS

RESISTANCES	LOCATION OF RESISTANCES											
	WEST WALL, W		EAST WALL, E		NORTH WALL, N		SOUTH WALL, S		WHITE ROOF, R <sub>1</sub>		GREEN ROOF, R <sub>2</sub>	
	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$	$\frac{F \text{ DEG}}{(BTU/HR)}$	$\frac{1000}{OHMS}$
$\frac{R_s}{4}$	0.00338	54.1	0.00315	50.4	0.00230	36.8	0.00600	96.0	0.00368	58.9	0.00303	48.5
$\frac{R_s}{2}$	0.00675	108	0.00631	101	0.00461	73.8	0.0120	192	0.00737	118	0.00607	97.2
$\frac{R_s + \frac{R_1}{6}}{4}$	0.0307	491	—	—	—	—	—	—	—	—	—	—
$\frac{R_1}{3}$	0.0545	871	—	—	—	—	—	—	—	—	—	—
$\frac{R_1 + \frac{R_p}{2}}{6}$	0.0296	474	—	—	—	—	—	—	—	—	—	—
$\frac{R_p}{2}$	0.00225	36.0	0.00210	33.6	0.00154	24.6	0.00401	64.1	—	—	—	—
$\frac{R_s + \frac{R_s}{6}}{4}$	—	—	0.00497	79.5	0.00363	58.0	0.00947	152	—	—	—	—
$\frac{R_p + \frac{R_s}{6}}{2}$	—	—	0.00392	62.7	0.00286	45.7	0.00748	120	—	—	—	—
$\frac{R_s}{3}$	—	—	0.00364	58.1	0.00266	42.5	0.00693	111	—	—	—	—
$\frac{R_d}{12}$	0.0318	556	0.0545	871	0.0223	356	0.0170	272	—	—	—	—
$\frac{R_d}{6}$	0.0697	1115	0.1091	1750	0.0445	712	0.0340	544	—	—	—	—





## APPENDIX C

## Radiation Exchange Network Within the Test House

Low temperature, long-wave radiation exchange occurs within any enclosure, such as the inside of the test house, between all surfaces and objects that "see" each other. A radiation exchange network accounting for direct radiation exchange and all inter-reflections between (m) surfaces will be derived on the basis of the following postulates:

1. All surfaces are opaque with respect to long-wave radiation.
2. All surfaces are diffuse emitters and reflectors.
3. The intervening media is non-absorbing.
4. All surfaces have a uniform temperature and radiosity.
5. The radiation properties of all surfaces are independent of wavelength over the spectrum considered (grey surfaces).

The net radiant energy exchange at a particular surface boundary (i) may be expressed as the difference in energy absorbed and emitted or as the difference in incident energy and radiant flux leaving the surface. Thus,

$$(q_{\text{net}})_i = A_i \epsilon_i (H_i - W_i) \quad \text{. . . . . (C-1)}$$

and

$$(q_{\text{net}})_i = A_i (H_i - R_i) \quad \text{. . . . . (C-2)}$$

Combining Equations C-1 and C-2 results in

$$(q_{\text{net}})_i = \frac{A_i \epsilon_i}{\rho_i} (R_i - W_i) \quad \text{. . . . . (C-3)}$$

Applying the resistance concept to Equation C-3 we obtain

$$(R_r)_i = \frac{R_i - W_i}{(q_{\text{net}})_i} = \frac{\rho_i}{A_i \epsilon_i} \quad \text{. . . . . (C-4)}$$

and

$$(q_{\text{net}})_i = \frac{R_i - W_i}{(R_r)_i} \quad \text{. . . . . (C-5)}$$

We may obtain another expression for the net radiant energy transfer at surface boundary (i) by determining the value of  $H_i A_i$  in Equation C-2.

$$H_i A_i = \sum_{n=1}^{n=m} R_n A_n F_{in} \quad \text{. . . . . (C-6)}$$

where

$$\sum_{n=1}^{n=m} R_n A_n F_{in} = \text{the summation of radiant flux leaving all surfaces that see surface (i) and is incident on surface (i).}$$

Then

$$(q_{\text{net}})_i = \sum_{n=1}^{n=m} [R_n - R_i] F_{in} A_i \quad \text{. . . . . (C-7)}$$

Again applying the resistance concept to Equation C-7 we obtain

$$(q_{\text{net}})_i = \sum_{n=1}^{n=m} \frac{R_n - R_i}{R_{in}} \quad \dots \quad (C-8)$$

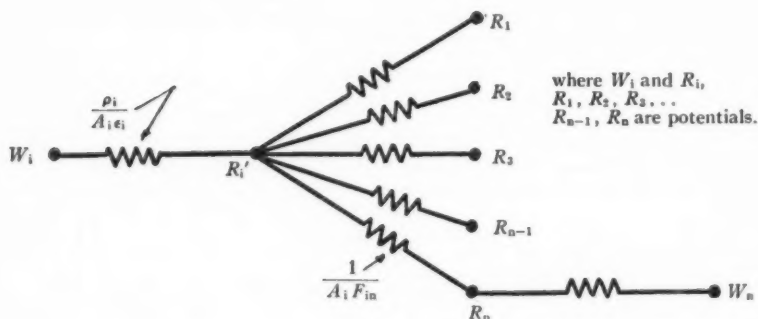
where

$$R_{in} = \frac{1}{F_{in} A_i} \quad \dots \quad (C-9)$$

also, by the law of reciprocity

$$F_{in} A_i = F_{ni} A_n \quad \dots \quad (C-10)$$

Note that Equations C-5 and C-8 describe an energy balance at a node in a resistance network. For example, radiation exchange at the  $i$ th surface may be represented by



An energy balance at  $R_1$  would result in

$$\frac{R_1 - W_i}{\frac{\rho_i}{A_i \epsilon_i}} = \sum_{n=1}^{n=m} \frac{R_n - R_1}{\frac{1}{A_i F_{in}}} \quad \dots \quad (C-11)$$

An equation similar to Equation C-11 may be written for each of the  $m$  surfaces resulting in a closed resistance network representing the system of ( $m$ ) radiating surfaces.

It is apparent that the potentials at each node of such a network are proportional to  $T^4$ . In order to insert such a network into the thermal circuit representing the conduction paths of the test house, it would be desirable to adjust the resistors so that a first power temperature potential at each of the ( $W$ ) nodes would result in the same heat or current flow through all of the resistors. To determine the proper adjustment to make in each resistor let us consider a path through the network between  $W_1$  and  $W_n$ .

Then

$$\frac{t_1 - t_n}{CR} = \frac{W_1 - W_n}{R}$$

where,

$R$  = sum of all resistors between  $W_1$  and  $W_n$ .

$C$  = the adjustment in resistance to linearize the potential.

Solving for  $C$  results in

$$C = \frac{t_i - t_n}{W_i - W_n} \quad \text{. . . . . (C-12)}$$

but,  $W_i = \epsilon_i \sigma T_i^4$  and  $W_n = \epsilon_n \sigma T_n^4$ . If the value of emissivity is the same for all surfaces

$$C = \frac{t_i - t_n}{\epsilon \sigma (T_i^4 - T_n^4)} \quad \text{. . . . . (C-13)}$$

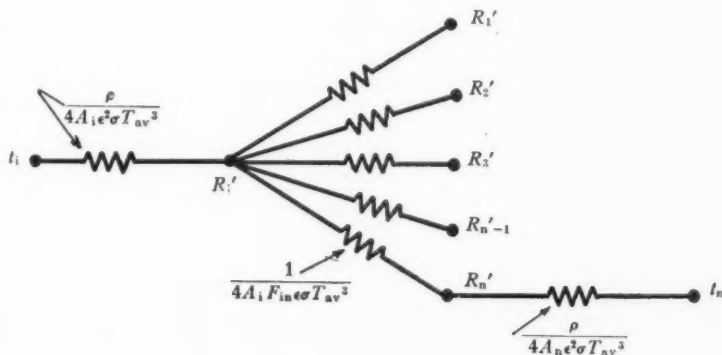
A good approximation for small differences in temperature is

$$T_i^4 - T_n^4 = 4(t_i - t_n)t_{av}^3.$$

Then

$$C = \frac{1}{4\epsilon \sigma T_{av}^3} \quad \text{. . . . . (C-14)}$$

and the network previously shown may be replaced by a network with linearized potentials.



Radiation exchange within the test house was represented by a similar resistance network applied to 8 surfaces and is shown in the thermal circuit diagram for the test house in Fig. 3.

The shape moduli required to evaluate the network resistors involved only finite rectangular areas contained either in parallel or perpendicular planes in accordance with dimensions given in the schematic sketch of Fig. 2. These factors were computed with the aid of equations and charts in references given below\*. An average surface temperature of 544 Rankin and an emissivity of 0.90 was used in the evaluation of the network resistances. A summary of the shape moduli and network resistances is given in Tables C-1 and C-2.

\* *Panel Heating and Cooling*, by B. F. Raber and F. W. Hutchinson (John Wiley & Sons, Inc., New York, 1947). *Radiant-Interchange Configuration Factors*, by D. C. Hamilton and W. R. Morgan (TN2836, NACA, December 1952).

TABLE C-1—SUMMARY OF SHAPE MODULI FOR INSIDE RADIATION EXCHANGE NETWORK

$F_{WE}$	0.100	$F_{EW}$	0.100	$F_{FW}$	0.135	$F_{NW}$	0.145
$F_{WN}$	0.215	$F_{EN}$	0.195	$F_{FE}$	0.135	$F_{NE}$	0.132
$F_{WG}$	0.155	$F_{EG}$	0.156	$F_{FG}$	0.083	$F_{NG}$	0.099
$F_{WR1}$	0.0604	$F_{ER1}$	0.059	$F_{FR1}$	0.121	$F_{NR1}$	0.101
$F_{WR2}$	0.089	$F_{ER2}$	0.089	$F_{FR2}$	0.157	$F_{NR2}$	0.180
$F_{WF}$	0.146	$F_{EF}$	0.146	$F_{FN}$	0.163	$F_{NF}$	0.080
$F_{WF}$	0.255	$F_{EF}$	0.255		0.204		0.260
TOTAL	1.02	TOTAL	1.00	TOTAL	1.00	TOTAL	1.00
$F_{R1W}$	0.104	$F_{R2W}$	0.141	$F_{RN}$	0.183	$F_{GN}$	0.221
$F_{R1E}$	0.104	$F_{R2E}$	0.141	$F_{RW}$	0.193	$F_{GW}$	0.089
$F_{R1N}$	0.312	$F_{R2N}$	0.115	$F_{RE}$	0.194	$F_{GE}$	0.148
$F_{R1G}$	0.032	$F_{R2G}$	0.207	$F_{RF}$	0.195	$F_{GF}$	0.338
$F_{R1F}$	0.062	$F_{R2F}$	0.088	$F_{R1R2}$	0.034	$F_{GR1}$	0.078
$F_{R1R2}$	0.000	$F_{R2R1}$	0.000	$F_{SR1}$	0.266	$F_{GR2}$	0.135
$F_{R1F}$	0.347	$F_{R2F}$	0.298	$F_{SG}$	0.000	$F_{GS}$	0.000
TOTAL	0.96	TOTAL	0.99	TOTAL	1.06	TOTAL	1.01

TABLE C-2—SUMMARY OF INSIDE RADIATION EXCHANGE NETWORK RESISTANCES

	F DEG (BTU/HR)	1000 OHMS		F DEG (BTU/HR)	1000 OHMS		F DEG (BTU/HR)	1000 OHMS
$(R_W)_r$	0.001307	20.9	$R_{WN}$	0.0546	875	$R_{NS}$	0.0803	1284
$(R_E)_r$	0.001307	20.9	$R_{WS}$	0.0759	1213	$R_{NG}$	0.0787	1260
$(R_N)_r$	0.000881	14.1	$R_{WG}$	0.1950	3120	$R_{NF}$	0.0386	617
$(R_G)_r$	0.001627	26.0	$R_{WF}$	0.0461	737	$R_{NR1}$	0.0441	705
$(R_G)_r$	0.001930	30.8	$R_{WR1}$	0.0805	1289	$R_{NR2}$	0.0994	1590
$(R_{R1})_r$	0.001529	24.4	$R_{WR2}$	0.1320	2110	$R_{EN}$	0.0601	962
$(R_{R2})_r$	0.001276	20.4	$R_{WE}$	0.1177	1880	$R_{ER1}$	0.1105	1770
$(R_F)_r$	0.000690	11.03	$R_{EF}$	0.0460	735	$R_{ER2}$	0.0815	1303
$R_{FS}$	0.0750	1200	$R_{EG}$	0.1173	1880	$R_{ES}$	0.0755	1208
$R_{SR1}$	0.43	6880	$R_{FG}$	0.0515	824	$R_{R1G}$	0.223	3570
$R_{SR2}$	0.0550	880	$R_{FR1}$	0.0396	634	$R_{R2G}$	0.129	2060
			$R_{FR2}$	0.0382	611			

## SUPPLEMENTARY NOMENCLATURE

- $F_{12}$  = shape modulus defined as the fraction of radiant energy leaving surface (1) and incident on (2), dimensionless.  
 $h$  = unit thermal conductance, Btu per (hour) (square foot) (Fahrenheit degree).  
 $k$  = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).  
 $l$  = length, feet.  
 $p$  = partial pressure, inches Hg.  
 $R$  = thermal resistance, Fahrenheit degree per (Btu per hour).  
 $\Delta T = t_1 - t_2$

- $V$  = volume, cubic feet.  
 $W$  = radiant flux density or radiant energy emitted per unit time, per unit area, Btu per (hour) (square foot).  
 $x$  = spatial coordinate in the direction of heat flow, feet.

#### Greek Letters

- $\beta$  = fraction of diffuse solar radiation  $\left(\frac{H_d}{H_n}\right)$ , dimensionless.  
 $\gamma$  = mass density, pounds per cubic foot.  
 $\delta_s$  = declination of the sun, degrees.  
 $\delta_n$  = declination of the normal to a surface, degrees.  
 $\theta_n$  = hour angle of the normal to a surface, degrees.  
 $\lambda$  = wave length, microns ( $\mu$ ).  
 $\rho$  = reflectivity, dimensionless.  
 $\tau$  = transmissivity, dimensionless.  
 $\phi$  = latitude.  
 $\mu$  = Microfarads

#### Subscripts

- $T$  = total.  
 $w$  = water vapor.

#### Superscript

- $n$  = noon.

#### Abbreviations

- sur = sunrise.  
 sus = sunset.  
 pot = potentiometer.

## DISCUSSION

S. F. GILMAN, Syracuse, N. Y. (WRITTEN): Considering all the variables involved, the analogue predictions are in reasonable agreement with the experimental results. The heat from the instruments is so great that there probably is a considerable net heat flow outward. This, together with an indoor temperature swing of 35 deg, is far from representative of a practical installation. It appears important that tests also be conducted with the room temperature controlled at a constant value. The additional studies planned do not appear to include such tests; it is suggested that cooling equipment be added for this important purpose.

With a 35 F indoor temperature swing, considerable heat could be temporarily stored in such places as the instrument casings and thereby introduce an unaccounted-for time lag. If the thermal storage capacity of the instruments is a significant amount, appropriate resistances and capacitances should be added to the equivalent electrical circuit. In Fig. 5, the solar energy absorbed by the green roof is much greater than that absorbed by the white roof. Could lateral heat flow from the green to white sections be of sufficient magnitude to warrant insulating their joining planes?

It would be of interest to know how many days of tests were conducted before this set of data was obtained. Our experience in Syracuse is that days on which the outdoor air temperature has the same value at the beginning and end of a 24-hr period are very rare indeed. The author has been fortunate in obtaining an outside temperature that is truly periodic. Moreover, the *measured* values in the illustrations also show remarkable periodicity. Could the curves of Fig. 7 be inadvertently interchanged? It

is noted that the analogue predictions are as much as 5 deg from periodic, whereas the *measured* curve is precisely periodic. I would expect the analogue to yield similar initial and final values.

It would have been of considerable value to calculate the load curve by available procedures in THE GUIDE and superimpose it on Fig. 11 for comparative purposes. In addition, I would like to urge the author to obtain the average hourly load from the sensible load curve for the structure. By introducing this into the analogue, it should be possible to determine the resultant variation in indoor temperature.

CARL F. KAYAN, New York, N. Y. (WRITTEN): Analysis of the different components in the chain of thermal links between heat source and receiver has in the past 15 years been reported in the Society, as one approach, with analogy as the tool. This has been by electrical analogy primarily, though to a lesser extent, also by hydraulic, and for both steady-state† and unsteady-state (transient)‡ conditions.

The present paper is a very commendable contribution in the analysis of a major group of structure components, particularly on a transient basis. It is refreshing to have this analysis matched by experimental determination of the load variation of the equivalent physical structure, and we should all be grateful for the real progress that this paper evidences. My compliments to the author.

I should like to add to the further understanding of this general problem and have the following comments.

Ultimately the heat received within a cooled enclosure must be absorbed by some form of refrigerating equipment. The resultant association of a load structure, with all that it entails, with a refrigerating plant, we might term a *Cooled-Structure Complex*. Analysis of the total system, involving not only the behavior of steady-state loads as well as transients, and likewise the performance characteristics of the refrigerating plant absorbing the loads, presents a still further advanced, yet more complicated problem. It must be remembered that the refrigerating plant itself is a complex arrangement of heat transfer components coupled to an energy transport device, or heat pump, such as the compressor, with all its operating characteristics added in.

A simplified version of a cooled-structure complex may be seen in Fig. A, and involves simulation of the transient load variation with time. This has been developed as an element in a resistance network, and as such parallels the perhaps better known resistance-capacitance arrangement. Use of the resistance network in the analogizer study facilitates the combination of the structure load with the refrigerating system. Simplifying assumptions, so often necessary in a pioneer development, are found expedient. Idealized performance for the sake of clarification is assumed, although corrections to account for the performance of real equipment may be made. The essential purpose is to develop further tools for electrical analogue operations.

Let us consider a cooled-structure complex consisting of an insulated structure with various heat loads, internal-space heat transfer equipment, and a refrigerating plant composed of its different essential elements. The heat load to be absorbed by the refrigerating plant consists of the wall leakage coupled with the infiltration load, an internal load of electrical power and lights, and the heat given off by a material mass brought in from the outside. The first 2 may be regarded as of steady-state nature (the wall assumed to be of negligible heat capacity), and the last, of unsteady-state (transient) nature and representing a load of warm material cooled transiently from the outside temperature down to the space temperature within the structure.

The total load to be absorbed by the refrigeration plant through the evaporator, plus the machine input energy to transport it to the higher condensing temperature, must

† Effect of Floor Slab on Building Structure Temperatures and Heat Flow, by C. F. Kayan (ASHVE TRANSACTIONS, Vol. 53, 1947, p. 377), and  
Electric Analogizer Studies on Panels with Imbedded Tubes, by C. F. Kayan (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 205).

‡ Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation and Convection, by C. S. Leopold (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 389).

ultimately be liberated in the plant condenser. The performance of the condenser depends on its heat-transfer conductance, the temperature difference between the condensing temperature and the coolant inlet temperature, and the coolant conditions of flow-rate and specific heat. Thus, for given conditions of condenser operation, the condensing temperature will depend on the hourly heat-flow rate to the condenser. In addition, the actual performance of the refrigerating system will depend on the compressor displacement, the effect of the necessary pressure difference on the volumetric efficiency, and the need of the condenser and the evaporator to adjust their operating temperatures to the imposed heat-transfer loads.

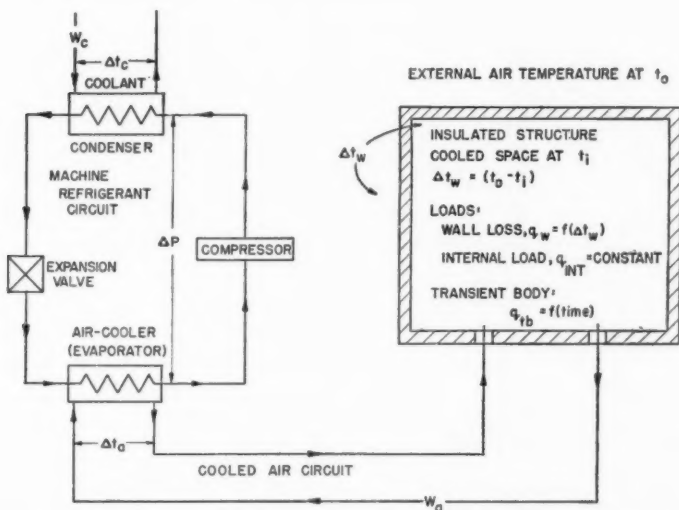


FIG. A. SIMPLIFIED VERSION OF A COOLED-STRUCTURE COMPLEX

It is apparent that the performance of this total plant system is a complex problem, and one difficult to forecast. The method of the electrical analogue analysis proves useful, in that the performance of different components may be interpreted in terms of their respective resistances and temperature differences. The basis for this approach was established in an earlier paper\*.

In the present analysis, the structure thermal circuit components that channel heat to the evaporator are superimposed on the refrigerating system. In addition to the steady-state loads, there is the transient hot-body load, and this has a temperature-time characteristic. This can be set up in terms of its initial temperature difference relative to the cooled-space temperature, and an equivalent transient heat-transfer resistance, such to give the liberation-energy flow rate as a function of time. Fig. B depicts this. Thus the total structure heat load to be dissipated to the evaporator (air-cooler) will additively consist of the wall plus infiltration load, the constant internal

\* Electrical Analogue Application to the Heat Pump Process, by C. F. Kayan (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 361).

load, and the product contribution (transient); collectively they can be shown as a function of time, somewhat similarly as shown in Fig. 11 of the author's paper. They sum up as a time-temperature variation for the evaporator heat load, which means that the internal space temperature will likewise vary with time, and this is illustrated in Fig. C.

To repeat, Mr. Buchberg's paper makes a real contribution in furthering the establishment of internal structure performance characteristics which in the future should be useful in the prediction of the performance of the overall system-complex of load plus its air conditioning plant.

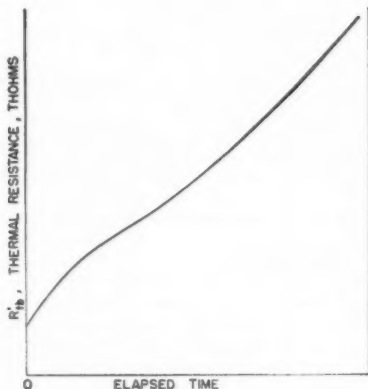


FIG. B. TYPICAL VARIATION OF TRANSIENT BODY RESISTANCE AS A FUNCTION OF TIME

H. T. GILKEY, Cleveland, Ohio, (WRITTEN): It is extremely interesting to see this report of an application of the thermal circuit analogy to an actual structure, and to learn of the excellent results shown by the comparison of predicted and measured results.

Unfortunately, the prediction of the diurnal variation of sensible air conditioning load necessary to maintain a constant air temperature (Fig. 11) is little more than an expansion of previous analyses. It does, however, provide an analysis of a building which has walls which are not homogeneous and which contain air spaces. Further, the study considers the loads for an entire building. The results indicate that the analogue is a powerful tool with this type of construction.

In the reporting of future work, it is hoped that the authors will be able to refrain from attaching more than one meaning to any one symbol or subscript. It is also to be hoped that such quantities as heat flux will be identified more precisely as to direction of flow, *i.e.* into or out of the building.

The wisdom of neglecting the reflection from the parapet north of the building cannot be questioned so far as the study reported is concerned. The effect of this reflection is shown in Fig. 9 and it would be advisable to reconsider the neglecting of this and certain other simplifying assumptions for studies in which the building was to be cooled. An indication of the importance of reflections from nearby surfaces was found in Warm Air Heating Research Residence No. 3 at the University of Illinois. Experience there showed maximum exterior surface temperatures on the north wall of from 4 to 10 deg higher than the outdoor air temperature at that time. Furthermore, these

maxima occurred before the occurrence of the maximum outdoor-air temperatures. This was attributed to reflection from a white frame house located approximately 25 ft north of the residence.

Other simplifications which might be reconsidered are that negligible infiltration exists and that existing moisture transfer is small. Both of these factors are virtual unknowns so far as heat gains are concerned. It would be a great contribution if, by use of the circuit analysis, we could evaluate both infiltration and moisture loads. There is reason to believe that evaporation of stored moisture contributes to the so called fly wheel effect, and comparisons of circuit analyses with experimental results could point the way to more accurate estimation of this effect.

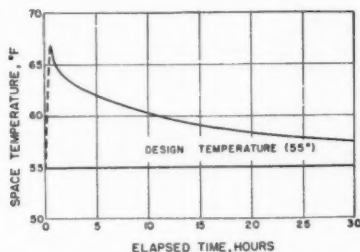


FIG. C. VARIATION OF COOLED-STRUCTURE SPACE TEMPERATURE AS A FUNCTION OF TIME

In addition to the future studies mentioned, it is to be hoped that consideration will be given to studies with a constant indoor-air temperature and to studies with a cooling unit of essentially constant capacity, such as those used in residential installations. In the case of the constant indoor-air temperature study, it should be possible for continuous records to be kept of the rate of heat removal from the space as well as to heat gains from instruments and similar sources. Furthermore, separation of latent from sensible heat removal should be possible in both cases.

PRESTON McNALL, Hopkins, Minn. (WRITTEN): The author has done valuable work in presenting the complete picture of this complicated thermal problem. I feel that the electric analogue presents the best possibility for the accurate and economical solution of the thermal storage problem. It is to be hoped that work of this type can establish the validity of the analogue design principles so that a larger program to obtain practical design information can be started. If the analogue validity can be established, then work can be undertaken without the necessity of checking it on full-scale models. When that point is realized, the vast variety of construction combination making up our homes, offices and factories can be analyzed to provide useful design information, not only for cooling but for heating or process work as well.

Our electric analogue of a house, heating system and control system is now nearing completion here at our Research Center and preliminary tests indicate that adequate accuracy can be obtained on fairly simple circuits representing wall sections and roofs. The simplicity and economy of working with a handful of resistors and condensers to get basic transient conditions on a wall section compared to testing a full-scale model is obvious.

The author states that more work should be done on reduction of circuit elements by simplification of *lumping*. If the thermal circuit equations are to be solved numerically, the further simplification is attractive. For an analogue designed for this type of work simplification is of much less economic importance. If the analogue is to be subjected to

different temperature frequencies loss of accuracy accompanies any further simplification. I would like to suggest future work on distributed RC transmission lines for analogue work rather than *lumped* elements.

F. R. O'BRIEN\*, Birmingham, Ala.: Mr. Buchberg has presented another example demonstrating the value and the power of the electric analogue method in the study of heat flow problems.

While the basic structure upon which he has based his problem is a simple one, the relationship of the factors considered is indeed complex; complex enough to make the direct computation of the effects of changing outside temperature and variable solar load a laborious task.

The possibilities of the analogue computer as a design tool are apparent. The inherent value of a device of this type lies in its ability to consider rapidly the effects of changing one or more boundary variables, and to present solutions in a manner easily interpreted.

Mr. Buchberg correctly points out that experimental studies of the thermal behavior of full scale systems are costly. He may well have added that they are usually inadequate in obtaining all the data that the experimenter finally realizes he needs for complete interpretation of his results. With the electric analogue the thermal circuit can be added to conveniently and quickly to furnish all the data that may be needed.

G. V. PARMELEE, Cleveland, Ohio (WRITTEN): Mr. Buchberg's excellent paper demonstrates again how complete thermal systems can be diagrammed and analyzed. It is a further step toward the goal of making the use of thermal-circuit techniques commonplace in analyzing the thermal behavior of buildings and in determining the proper capacity and responsiveness of cooling and heating systems.

When Dr. Nottage first proposed the application of thermal circuit techniques to load estimating and other problems of interest to air-conditioning engineers, some people confused, and perhaps still do, the thermal-circuit *concept* with the electrical analogue *method* of solving thermal circuits. Therefore, I think it is worthwhile at this time to emphasize that the thermal circuit shown in Fig. 3 of the paper diagrams a thermal system consisting of an entire building, with its thermal resistances and capacitances, interconnected with its environment and various heat sources and heat sinks. In this particular case the figure is also the circuit diagram of the tool used to solve it, namely an electrical analogue.

Some of the idealizations made in representing the real system by the circuit diagram are subject to discussion. However, Mr. Buchberg's work demonstrates a particular advantage, among others, of the electrical analogue method of solving circuit problems in that a thermal system can be represented in considerable detail without making the circuit become more difficult to solve, as would be true if some other methods were used. For example, the non-homogeneous wall construction has been represented in good detail. Other work to be reported is of practical importance in that it will show to what extent simplifications can be made consistent with good engineering in the results. This should provide guidance in setting up other problems.

The importance of taking into account the effect of all of the heat-storing capacity of structures and their contents in designing cooling—and heating—systems is now well known. Except in special cases, however, the circuit analysis technique, which provides the means of doing this, is too time taking for general use. Hence, systematic series of problems need to be solved from which practical-use engineering data can be derived. The economic importance of this is such that this phase of the Society's research program should be activated as soon as possible. Strong support from the air-conditioning industry is essential to carrying out such a program. From the experi-

\* Director, Southern Research Institute.

ence and reports to date of workers in the field of thermal circuits, it now seems clear that the electrical analogue method of solving thermal-circuit problems would be most economical of time and money in such a program.

I have several comments on the paper itself as follows:

1. With respect to each set of plotted points in Fig. 5 and the points for the outside air temperature curve of Fig. 10, the figure captions should indicate that these are the input currents and potentials which were fed into the circuit, not analogue predictions. The departures of the solar radiation inputs from the observed values may be responsible for some of the differences between computed and observed inside air temperatures.

2. One can only speculate on the effect of omitting an input current for the diffuse solar radiation transmitted through the large south-facing window. Though reduced by shading, this input should have a significant effect on the computed indoor air temperature.

3. It is noted that the power input to the recording apparatus is a large constant factor. Hence, the actual influence of the periodic variation in sun intensity and air temperature is, to a considerable extent, hidden in the solutions given in the paper.

4. Mr Buchberg points out that the thermal circuit solution can be only as good as the evaluation of the circuit parameters and boundary conditions. This evaluation is far from a simple matter and needs very careful attention in future comparisons between computed and observed results. Comparison of predicted and observed thermal behavior such as the one Mr. Buchberg has made serves two important purposes. First, they demonstrate where better data is needed on, or understanding of, heat transfer, air distribution, mixing and infiltration. Second, successful comparisons of at least a few representative cases are essential to general acceptance of computer results by air-conditioning engineers. It is evident, therefore, that such comparisons are a necessary part of a comprehensive research program on thermal circuits.

The results of further work in progress and planned are awaited with great interest.

EVERETT PALMATIER, Syracuse, New York: I would like to direct my remarks specifically to Fig. 10. Before doing so, let me say that I have made such remarks at past meetings when papers were given on periodic heat flow research. Note from Fig. 10, that the outside temperature variation during the day is roughly 20 deg. The inside temperature variation is 30 deg. Also, observe that the temperature curves are roughly in phase. One of the reasons for this in-phase relation of the temperature curves and the fact that the inside temperature variation is so great, is that most of these periodic heat flow studies have no thermal capacitance within the shell of the space. In the analogue used in these experiments there was a small capacitance representing the capacitance of air. In practical structures there is a considerable amount of material within the shell through which the periodic heat flow takes place. When one notes the relation between such curves in practical cases, it will be found that the inside temperature variation is a very small fraction of the outside temperature variation, even with substantially larger direct solar gains than are included in this paper. Also it will be found that the phase difference between maximum outside and inside temperatures is very often as much as 5 or 6 hours.

**AUTHOR'S CLOSURE:** It is indeed gratifying to see so much interest displayed in a paper dealing with the application of circuit analysis to the prediction of cooling and heating loads. Regardless of what method of solution is used in solving the final circuit, representation of complex heat and mass transfer systems by thermal circuits makes it possible with a systematic procedure to estimate the importance of many factors and variables in predicting the performance of the system. Fortunately, useful engineering results can be obtained in spite of the fact that many approximations are made in arriving at a final solution of the circuit. This is due to the large number of factors present which influence the behavior of the system significantly but in a small way. It is true that any very large influence must be accurately represented to obtain useful results.

Several interesting questions have been raised by Mr. Gilman. Regarding the large indoor temperature swing, which certainly is not representative of an acceptable installation, it should be mentioned that this condition was purposely sought in order to make temperature-time comparisons between the actual system and the analog prediction more meaningful. In this investigation the thermal capacity associated with the instruments located in the space was considered to be negligible. Additional solutions of the thermal circuit have been obtained for the case where the indoor air temperature was maintained constant and will be reported in a future paper. Experimental studies with the use of cooling equipment to maintain the space temperature constant have not been planned because of insufficient funds.

Some question was raised about the lateral heat flow between the white and green roof sections. It was felt that for purposes of analysis unidirectional heat flow was justified on the basis that the resistance to heat flow in the lateral direction was large compared to the direction perpendicular to the long dimension. Insulation in a joining plane between the two roof areas would not have served any useful purpose.

Concerning the periodicity of the measured and predicted temperatures, Mr. Gilman is correct in expecting the analogue predictions to be precisely periodic, inasmuch as the inputs were all periodic. Examination of the tabulated data given in the original report to the Society indicates a maximum deviation from exactly periodic behavior of 2.8 deg F rather than the 5 deg F mentioned. This resulted from some difficulty in interpreting the film record of potentials at discrete time intervals. This difficulty has now been overcome by making continuous recordings of the potentials of interest and simultaneously recording a timing signal originating from the outside air temperature input generator.

Before the experimental data were obtained, weather conditions were observed for several months until a period of cloudless days appeared early in September. Actual measurements were taken for several days. The 24-hr period beginning September 8 was selected on the basis of periodicity.

With respect to Mr. Palmatier's remarks, I might say that I agree with him that additional capacity in the space will tend to reduce indoor temperature swings. Further if by a practical structure he means one having much more mass such as concrete or masonry construction, rather than conventional wood frame construction used extensively in Southern California, then I agree that with his practical structure one would expect less inside temperature variation and much greater phase difference between maximum outside and inside temperatures. The important point is that thermal circuit analysis, demonstrated for a particular system in this investigation, is readily adaptable to other systems having different values of thermal resistance and capacitance.

Mr. Gilkey has suggested several refinements in the analysis for consideration in future studies. Reflections from adjacent buildings as well as the effect of shading devices can be introduced into the solar input circuit. To accomplish this, Equation A-13 for incident solar radiation (see Appendix A) might be modified by multiplying by a function  $\psi'(\theta)$  to give

$$(H_D)_a = [A + B \cos(\theta_a - 15\theta)]\psi'(\theta)$$

A similar function  $\psi(\theta)$  was used to account for sunrise and sunset. Infiltration can be accounted for in the circuit by inserting a resistance between the inside air temperature and outside air temperature such that the heat flow through the resistance is  $Wc_p(\tau_o - \tau_i)$ . The resistance would then equal  $\frac{1}{Wc_p}$  where  $W$  is the infiltration rate and  $c_p$  is the unit heat capacity of air. For the case where  $W$  is time variable it might be more convenient to use a ( $\pm$ ) current source in the analogous circuit. Moisture loads resulting from the evaporation of water from a surface and diffusion into the

space could also be treated as a current source and computed from the equation  $q = Ah_m L(p_s - p_\infty)$  where  $A$  is surface area,  $h_m$  is a unit mass conductance,  $L$  is the heat of vaporization,  $p_s$  is the vapor pressure of water at the surface temperature, and  $p_\infty$  is the partial pressure of the vapor in the space. It should be recognized that the circuit representation offers no assistance in evaluating  $h_m$ .

Mr. McNall makes the interesting suggestion for future work to consider distributed RC transmission lines for analogue work rather than lumped element circuits. Discovery of practical distributed RC transmission elements would make possible the solution of problems that require too fine a mesh using a lumped parameter network. However for this application it is not clear why there should be any concern over the lumped element representation. Values of thermal diffusivity for the usual materials of construction are such as to make relatively few lumps practical and few input harmonics necessary.

With respect to simplification of network representation, solutions by means of electric analogue are not necessarily the most economical approach for every design problem. It appears worthwhile to study simplification of circuits with the objective of setting up hand computational procedures based on a classification of systems and a tabulation of complex impedances characteristic of typical structural components and determined with the use of an electric network computer in advance.

I would like to thank Professor Kayan for his kind remarks and emphasis on the importance of considering the whole system, including the air conditioning equipment. Professor Kayan has been doing a pioneering job in this area and we hope in the future to include consideration of equipment characteristics and control in our studies.

Mr. Parmelee correctly questions the interpretation of computed points given in Figs. 5 and 10. The computed points in Fig. 5 and also in Fig. 6 represent inputs. In Fig. 10 the computed points for the inside air temperature express the prediction and for the outside the input.

In connection with Mr. Parmelee's second comment, I can now report on the effect of including the transmission of diffuse solar radiation through the south-facing glass area. For the case where the inside air temperature is considered to be a dependent variable, the predicted space temperature was increased slightly during the day reaching a maximum difference of 1.9 deg at approximately 2:00 p.m. For the case where the space temperature was maintained constant, the increase in the peak sensible air conditioning load was approximately 8 percent.

It is true as Mr. Parmelee points out that the heat source due to the presence of recording equipment in the enclosed space was a potent factor in establishing the space temperature. However, it is not correct to conclude that the effect of periodic variations in solar irradiation and air temperature are hidden in the solutions. In the analysis of linear circuits it is convenient to separately consider the effect of d-c and a-c inputs based on the principle of superposition. On this basis it can be seen that the effect of large d-c inputs, such as the space heat source, is simply to raise the d-c level of the prediction without affecting or hiding the influence of the a-c inputs. Also, sensitivity in measuring the variations is preserved if the reference or ground potential is made equal to the d-c level.

We now have available solutions of the circuit for a constant space temperature giving the quantitative contributions of all of the inputs in terms of sensible air-conditioning load curves. These will be presented in a future paper.

In conclusion I would like to say that H. B. Nottage made possible these studies through his initiative, encouragement, and advice.

## SYMBOLS

	Source of constant potential
	Source of periodically varying potential
	Current source
	Amplifier
	Thermal resistance also Fixed resistor
	Variable or adjustable resistor
	Potentiometer or voltage divider
	Double throw relay
	Current Generator
	Wires connected
	Wires crossing (not connected)
	Ground
	Twin Diode



**1544**

## A METHOD FOR DETERMINING WINTER DESIGN TEMPERATURES

By M. K. THOMAS\*, TORONTO, ONT., CANADA

A WINTER design temperature may be broadly defined as the coldest temperature which is likely to recur frequently enough during the average winter to justify its use in the design of heating systems for structures. These temperatures are used in calculating the heat loss which such systems may be called upon to overcome in normal operation.

While winter design temperatures have been used for some time in Canada, no one has as yet published a method by which these temperatures may be selected on a systematic basis across the country. In the past some designers have used values obtained by adjusting the extreme or average minimum temperatures according to a rule, while others have estimated values directly. In the preparation of winter design temperature charts for the Climate Part of the revised National Building Code of Canada, it was necessary to devise a logical method for obtaining these temperatures and to give a specific definition to a Canadian winter design temperature.

At the first meeting of the Technical Committee on Climate for the National Building Code, it was decided to select design temperatures on the bases of four probabilities (1, 2½, 5 and 10 percent) so that the designer might have a choice according to the heat capacity and the use of a building. Considering the volume of calculations involved in obtaining these design temperatures, it was also decided to base them on conditions during the coldest month of the year only. Accordingly, the following definition was adopted: *Winter design temperatures for bases of 1, 2½, 5 and 10 percent are the temperature values expressed in Fahrenheit degrees at or below which 1, 2½, 5 and 10 percent of the January hourly outdoor temperatures occur.*

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\* Deputy Superintendent, Climatological Services, Meteorological Division, Department of Transport.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

## METHOD

January is usually the coldest month of the year throughout most of Canada†. Normal January temperatures range from  $+10$  to  $+25$  deg F in the more densely settled portion of Eastern Canada, from  $-5$  to  $+15$  deg F in the southern Prairie Provinces and from  $+15$  to  $+35$  deg F in settled British Columbia. While a longer period of time would have been desirable, the 10 Januaries from 1941 to 1950 inclusive, were chosen as a basis for this study. However, the period proved to be fairly representative, as the departures from normal of the mean January

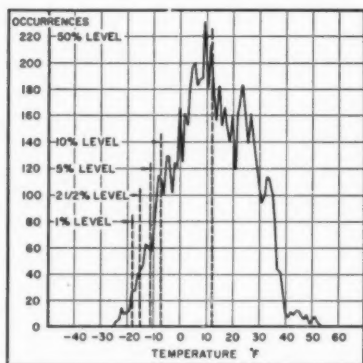


FIG. 1. JANUARY HOURLY TEMPERATURE FREQUENCIES AT OTTAWA, ONT., FOR THE PERIOD 1941-1950. TEMPERATURE FREQUENCIES HAVE BEEN ACCUMULATED FROM THE COLDEST TEMPERATURE UPWARD TO INDICATE THE 1, 2½, 5, 10, AND 50 PERCENT LEVELS

temperature during this period were mostly within the range  $-1$  to  $+2$  deg in settled Southern Canada and  $+4$  to  $-2$  deg in Northern Canada. The largest departures from the long term normal occurred in the sparsely settled Mackenzie Valley and Yukon Territory.

The first step was to analyze the hourly temperature frequencies from 10 representative locations. The hourly temperature records for 100,000 observations were transferred to punched cards for machine analysis, from which it became apparent that more information was needed. Hourly temperature frequencies for an additional 25 stations were then obtained by clerical methods for the coldest 10 percent of the hours.

† February averages colder only in Southern Ontario, Nova Scotia and New Foundland where the difference varies from 0 to  $-2$  deg and in the Arctic Islands where the differences are 0 to  $-6$  deg. Most differences, except in the Arctic, are 1 deg or less.

To obtain design temperatures for the desired (or any) percentage level at these 35 stations, it was necessary to accumulate the temperature frequencies from the lowest temperature up to the desired percentage level. The frequency distribution of 7,440 January hourly temperatures at Ottawa is shown in Fig. 1. The position of the 74th lowest temperature is indicated as the 1 percent level, the 186th as the 2½ percent level, the 372nd as the 5 percent level and the 744th as the 10 percent level. The 50 percent level and the mean January temperature for the same 1941-1950 period are also shown.

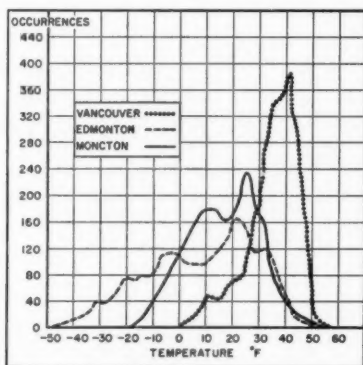


FIG. 2. JANUARY HOURLY TEMPERATURE FREQUENCIES AT VANCOUVER, B. C., EDMONTON, ALTA., AND MONCTON, N. B., FOR THE PERIOD 1941-1950. THE CURVES HAVE BEEN SMOOTHED

In order to obtain a more complete coverage of the country, a member of the National Building Code Committee on Climate, H.C.S. Thom of the U. S. Weather Bureau suggested the use of a relationship between (1) the variability of January monthly mean temperatures, and (2) the difference between the monthly mean temperature and the design temperature. As Canada is divided into several regions according to the type of temperature frequency distribution, the use of several regional relationships was suggested. Fig. 2 illustrates the different types of temperature distribution and indicates the need for regional relationships. The Vancouver frequency distribution illustrates the compact distribution of hourly temperatures typical of the maritime climate along the Pacific Coast which was also found to exist in the Great Lakes region of southern Ontario. The broad Edmonton distribution is representative of the continental climate in Western Canada, while the intermediate Moncton distribution is typical of stations in Eastern Canada. Fig. 3 shows the different regions into which the country was divided along with the location of the stations for which design temperature values were obtained.

To obtain these regional relationships, monthly mean temperatures and standard deviations were calculated from the 35 locations for the 10-year period 1941-1950. Standard deviations in Southern Canada were about 3 deg along the Atlantic Coast and increased steadily to 13 deg in Alberta and interior British Columbia and then decreased to 5 deg along the Pacific Coast. In Northern Canada the values exceeded 8 deg in the Eastern Arctic decreasing to 4 deg in the Western Arctic.

The standard deviation value for each station was plotted against the difference between the mean temperature and the known  $2\frac{1}{2}$  percent design temperature. These plots and the resulting lines of best fit are shown in Fig. 4. An excellent

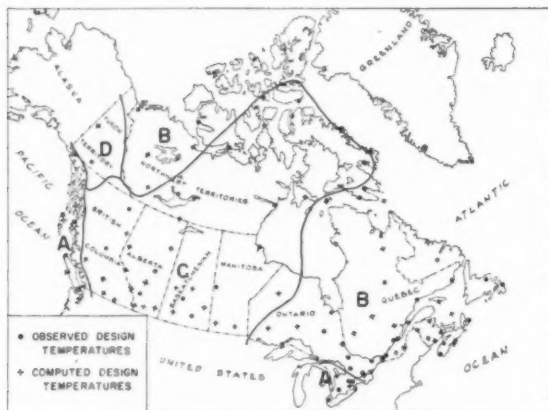


FIG. 3. ZONES OF REGIONAL RELATIONSHIPS AND LOCATION OF STATIONS FROM WHICH DATA WERE USED

relationship was apparent for the Western Canada stations. The three plots which do not lie along the line (C) represent stations on the edge of this region. The Eastern Canada plots were more scattered than the first group but are represented fairly well by line (B). Although the method of least squares was used to obtain the original relationship lines it was found that satisfactory lines could be drawn by inspection of the plotted points. The values for the Maritime stations were bunched and so the line (A) was given the same slope as (B). Similarly a line (D) was drawn through the single Yukon stations which was parallel to the relationship line (C).

To utilize these relationship lines, January mean temperatures and standard deviations were calculated for an additional 80 stations for the period 1941-1950. Knowing the standard deviation and using the proper regional relationship line, the difference was read off between the mean temperature and the  $2\frac{1}{2}$  percent design temperature. Since the mean temperature was also known, the calculated design temperature was obtained by subtracting the previously mentioned dif-

ference from the mean temperature. The data from stations near the boundaries of different zones were adjusted accordingly as were data at stations with known unusual climatological characteristics. The design temperature values were plotted and the chart shown in Fig. 5 was drawn. In settled Southern Canada these values range from  $-35$  deg in the interior to  $0$  deg on the Atlantic Coast and  $10$  deg on the Pacific Coast.

Similar procedures could have been used to obtain relationships for the 1, 5 and 10 percent levels. This was attempted but it was found to be just as accurate if values were estimated using, with necessary adjustments, the basic relationships already found. In each region there were several locations where the exact fre-

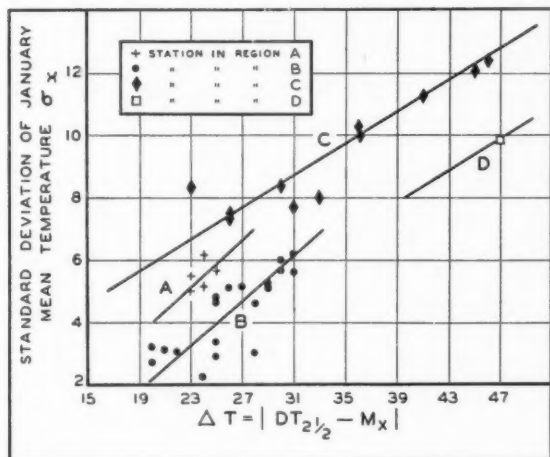


FIG. 4. REGIONAL RELATIONSHIPS BETWEEN STANDARD DEVIATION ( $\sigma_x$ ) AND THE DIFFERENCE BETWEEN THE MEAN TEMPERATURE AND THE WINTER DESIGN TEMPERATURE— $2\frac{1}{2}$  PERCENT BASIS

quencies were known for the lowest 10 percent of the temperatures. These observed design temperatures were used as reference points in estimating values for the other eighty locations. The resulting charts are similar in appearance to the  $2\frac{1}{2}$  percent basis chart and are not shown here.

#### COMPARISON OF CITY AND AIRPORT DESIGN TEMPERATURES

As hourly temperature data were available for city and airport stations at Toronto and Montreal, winter design temperatures were obtained for these stations and are shown in Table 1. As a result of the urban influence, city temperatures

in the winter are usually slightly higher than those at adjacent airports. Since these 2 cities are the largest metropolitan areas in Canada, large city-airport differences are not expected in other urban areas. At Toronto and Montreal, differences in design temperatures between city and airport sites are about one degree greater than the differences between mean temperatures.

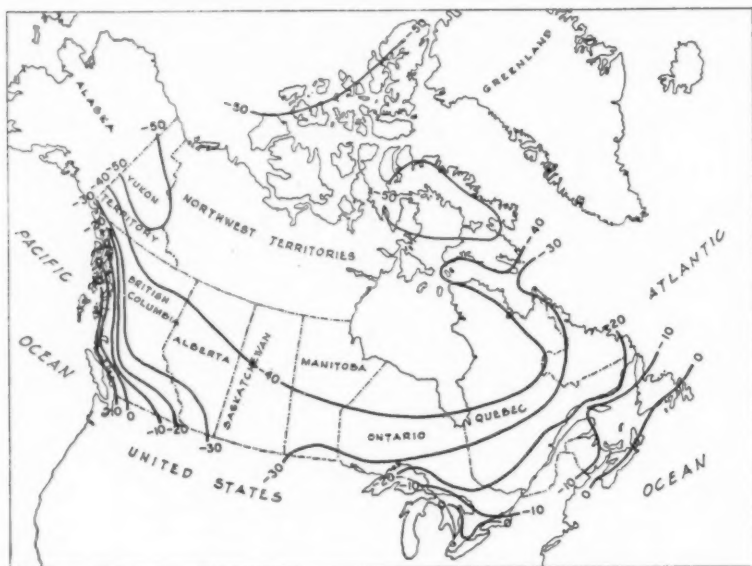


FIG. 5. WINTER DESIGN TEMPERATURE— $2\frac{1}{2}$  PERCENT BASIS FAHRENHEIT DEGREES

#### COMPARISON OF TAC $97\frac{1}{2}$ PERCENT BASIS AND CANADIAN $2\frac{1}{2}$ PERCENT BASIS WINTER DESIGN TEMPERATURES

Many readers will be interested in seeing how the design temperatures as obtained by this Canadian method compare with design temperatures selected by the ASHAE Technical Advisory Committee  $97\frac{1}{2}$  percent basis. This latter temperature is defined as *the hourly out-door temperature which has been equalled or exceeded  $97\frac{1}{2}$  percent of the total hours in December, January and February for the period of record.* The major difference is that the TAC method uses the three winter months instead of the single coldest month used in this paper. A minor difference in procedure is that the TAC method accumulates the temperature frequencies from the highest temperature down, while the method described here accumulates from the lowest temperature upward. A difference of 3 deg was found between the two methods at Toronto. The TAC  $97\frac{1}{2}$  percent value, based

on December, January and February is  $+3$  deg, while the present method gives a value of 0 deg on the  $2\frac{1}{2}$  percent level based on January temperatures alone.

TABLE 1—COMPARISON OF CITY AND AIRPORT WINTER DESIGN TEMPERATURES AT TORONTO AND MONTREAL (FAHRENHEIT DEGREES)

	JANUARY 1941-1950		DESIGN TEMPERATURES—PERCENT			
	MEAN TEMPERATURE	STANDARD DEVIATION	1	$2\frac{1}{2}$	5	10
Toronto						
City.....	24.5	5.2	- 4	0	5	10
Airport.....	21.2	5.7	- 9	- 4	1	6
Difference.....	3.3	-0.5	5	4	4	4
Montreal						
City.....	16.1	4.7	-12	- 9	-6	- 2
Airport.....	14.9	5.1	-14	-11	-8	- 4
Difference.....	1.2	-0.4	2	2	2	2

#### SUMMARY OF WINTER DESIGN AND RELATED TEMPERATURES

In summary, Table 2 is a tabulation which lists for 15 Canadian cities various winter design, mean and extreme temperatures. The Canadian design temperatures are based on the period 1941-1950 and where indicated are derived from airport temperature values. The standard temperature values are from 30-year records which are either entirely city records or a fairly homogeneous combination of city and airport records.

Study of the tabulation reveals that apparently the design temperatures in common use have been selected with the mean annual minimum temperature in mind since there is close correlation. These temperatures in common use, while never as low as the 30-year extreme minimum temperature, are usually lower than both the 1 and  $2\frac{1}{2}$  percent basis design temperatures as selected in this study.

#### ACKNOWLEDGMENTS

This paper records work done by the author in connection with the revision of the National Building Code of Canada. This was carried out as part of the author's duties as Climatologist to the Division of Building Research of the National Research Council (1951-1953), to which position he was seconded from the Meteorological Division, Department of Transport. The paper is published with the approval of Andrew Thomson, Controller of the Meteorological Division and of R. F. Legget, Director of the Division of Building Research, under whose immediate direction the project was undertaken. The Central Mortgage and Housing Corp., Ottawa, cooperated by transferring temperature data to cards for machine analysis.

TABLE 2—WINTER DESIGN AND COMPARATIVE TEMPERATURES AT 15 LOCATIONS IN CANADA (FAHRENHEIT DEGREES)

STATION <sup>a</sup>	WINTER DESIGN TEMP <sup>b</sup>				DESIGN TEMP IN COMMON USE <sup>c</sup>	30-YEAR STANDARD VALUES		
	PERCENT					MEAN JANUARY TEMP <sup>d</sup>	MEAN ANNUAL MINIMUM <sup>e</sup>	EXTREME MINIMUM <sup>f</sup>
	1	2½	5	10				
Vancouver (A), B.C.....	8	11	15	21	10	36	13	0
Edmonton (A), Alta.....	-39	-33	-29	-21	-40	8	-39	-55
Regina (A), Sask.....	-39	-34	-30	-25	—	2	-39	-54
Churchill (A), Man.....	-43	-42	-40	-37	—	-16	-43	-50
Winnipeg (A), Man.....	-33	-29	-25	-21	-35	1	-35	-44
Ottawa (A), Ont.....	-18	-15	-11	-7	-20	12	-26	-38
Toronto, Ont.....	-4	0	5	10	-10	24	-7	-22
Montreal, Que.....	-12	-9	-6	-2	-15	15	-16	-29
Quebec, Que.....	-16	-12	-9	-4	-20	12	-19	-32
Saint John, N.B.....	-6	-3	0	5	—	20	-12	-21
Halifax, N.S.....	0	4	7	11	—	32	-6	-21
Charlottetown, P.E.I.....	-6	-3	0	4	-10	19	-11	-23
St. John's (A), Nfld.....	-1	1	4	7	-10	24	-2	-10
Goose Bay (A), Nfld.....	-29	-26	-24	-20	—	0	-30	-35
Dawson, Y.T.....	-62	-56	-50	-39	-45	-16	-54	-73

<sup>a</sup> (A) indicates that observations were taken at the airport for the 10-year period 1941-1950.<sup>b</sup> Based on January 1941-1950.<sup>c</sup> HEATING, VENTILATING, AIR CONDITIONING GUIDE 1954, Chapter 12, p. 245 published by AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS, New York, N. Y.)<sup>d, e, f</sup> Thirty-year standard values are based on the period 1921-1950 and are either entirely city records or a fairly homogeneous combination of city and airport. The periods at Churchill, St. John's and Goose Bay are less than 30 years.

## DISCUSSION

S. A. HEIDER, Washington, D. C. (WRITTEN): The important feature of the new data described by Mr. Thomas, in my opinion, is the multiple choice of design temperatures which are based on different frequencies of occurrence. This permits the designer to choose a temperature suited to a particular job with regard to type of building, local exposure, or importance of an adequate heating system. It also places the responsibility for choosing a design temperature on the designer where it ought to be. He alone knows all the conditions of his particular job, and is the only person in a position to make a decision.

The choice of values given in Table 2 seems to begin in the range of the old temperature-in-common-use and extends upward to higher values of temperature. I would like to suggest that at least one set of values below the 1 percent range should be added to permit a choice of lower temperatures. This would take care of cities where the new 1 percent value has deviated sharply upward from the old temperature-in-common-use. It would permit a designer to use a lower and safer figure for his jobs until he has made analyses of old jobs to assure himself that the higher figures are really quite adequate. I personally believe that the 10 percent figures are generally much too high and that designers will be very reluctant to use them at all.

The method used by ASHAE's present TAC on Weather Data for deriving a comparable set of winter design temperatures, differs very much from that described by Mr. Thomas. This, perhaps, is not significant because the method used is not especially important as long as a set of multiple temperatures are provided for the designer to choose from. It would be interesting nevertheless to compute data for border cities by

both methods to compare the results obtained by the two methods. It may be that the results of both methods are comparable for all practical purposes, and that the only significant difference lies in the range of choice of design temperature in relation to the old temperatures-in-common-use.

H. E. DEGLER, Kansas City, Mo. (WRITTEN): This is an excellent paper on *Winter Design Temperatures* with Table 2 and Fig. 5 containing the summarized recommended values for Canada in convenient form. I am glad to note that Mr. Thomas used temperatures for the period 1941-50, which was a cold weather cycle. This opinion is based upon my belief in the Hale 11-year hot and 11-year cold weather cycle and an article by I. R. Tannehill, Assistant Chief, U. S. Weather Bureau, published in the September 1954 issue of *Country Gentleman*.

Not many years ago textbooks told us to find the winter design temperature by adding 10 degrees to the lowest recorded temperature that occurred during the previous 10 years. Adding 10F to column 8 in Table 2 would undoubtedly be too low as a design temperature for most cities when compared to the recommended column 2 which is more realistic. (However, adding 10 degrees to the *mean annual minimum* in column 7 would for most Canadian cities approximate Mr. Thomas's recommended values in column 2). The agreement of the Canadian method using only January temperatures compares favorably with the presently published ASHAE 97½ percent basis for the 3 winter months.

From a similar study (now in progress) made for USA cities by H. C. S. Thom of the Society's Technical Advisory Committee on Weather Data, the preliminary report indicates the probability of recommending the use of 92½ percent for USA cities instead of the current 97½ percent, which latter figure is now being recommended for Canada by Mr. Thomas. I wish that Mr. Thomas would comment on this from the standpoint of difference in climatic conditions which may justify the use of both values (92½ for USA and 97½ percent for Canada).

JOHN EVERETTS, JR., Philadelphia, Pa.: The data presented are, obviously, most important in any air conditioning or heating design and the TAC on Weather Data has been working, not only thru the U. S. Weather Bureau but also thru the Canadian Climatological Service with Mr. Thomas who is a member of our committee. I wish to say he has done excellent work, not only in the winter design temperature field, on which information has been presented in the paper, but also in the weather atlas which has been prepared for Canada.

In the various discussions there seems to be a feeling that both Mr. Thomas and our own committee are trying to set up a definite standard of temperature which must be used for design and I would like to clarify this point concerning our work and that of Mr. Thomas. These temperatures which he has set up and temperatures which we are setting up are temperatures from which engineers may select the one for a particular problem. For a light residential structure, one design temperature might be used; but for a heavily constructed office building which operates during the daytime only, one may use a higher temperature in order to reduce the costs of the installation.

One other thing I would like to point out is that most of these temperatures are macroclimatological conditions which cover an area and not microclimatological conditions which cover small localities within a given area. For example, for Toronto there are records where, at the edge of the lake, the temperature would be 0 F, and, about 5 miles in back of the lake, within the same hour, the temperature would be -15 F. It is not possible, at the moment, to detail the microclimatological data as some people would like to have done. In Washington, D. C. there may be a difference of 12 deg between the northeast and southwest districts within the same hour. The committee is trying to establish the probabilities of temperatures as they exist, and let the engineer in a given locality or a given microclimatic area decide for himself what temperature he should use to establish a proper condition for a given structure.

I want to compliment Mr. Thomas for the work he has done, not only on this paper, but also on the atlas. As far as the data in the GUIDE are concerned, we recognize that

they do not comprise data that we now recognize as being correct, and that will be changed, it is hoped, within the next year, on the basis of the studies now being made. For example, Boston, New York, Philadelphia, and Washington all now show a design winter temperature of 0 F. That is obviously not correct. On the basis of the data that the Weather Bureau is now working up on the probabilities that exist for the period in which temperatures have been recorded, Boston is shown as 0, New York as +5, Philadelphia +6 and Washington +10, which seems to be more in line with the geographical location of those areas and I think, within the next year more accurate data will be available for use.

**AUTHOR'S CLOSURE:** A set of values below the 1 percent level could be easily obtained from the data already compiled. In southern Canada the 0.5 percent level values would be about 3 deg lower than the 1 percent level values.

There is a comparison in the paper between the present TAC 97½ percent values and the 2½ percent values obtained. A comparison with the new proposed method using the lowest daily average temperature over a 30-year period is not possible at present since Mr. Thom's method of obtaining probabilities has not yet been published.

The period 1941-1950 seemed fairly representative as most of settled Canada experienced January temperatures that were within  $\pm 2$  deg of the long-term normal. An exception was southern Manitoba and Saskatchewan where departures were of the order of +3 deg from the long-term January normal.

Perhaps the extreme microclimatic conditions have been over-emphasized in considering the temperature regimes of cities. In Toronto there are occasions when large differences do occur between the lake front and the suburbs which are miles away and several hundred feet above the lake. However, on an average basis there is only a 3 to 4 deg temperature drop from the lake front to the interior.



**1545**

## PERIODIC HEAT FLOW THROUGH FLAT ROOFS

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**This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.**

**F**OR MANY YEARS engineers have been faced with the difficulty of properly evaluating the portion of the cooling load due to periodic heat flow through multi-layer building sections. One example of considerable importance is estimating heat gain through roofs of large single-story buildings, such as shopping centers, schools and industrial plants. An analytical solution to the problem was developed in 1946 by C. O. Mackey and L. T. Wright, Jr.<sup>1</sup> of Cornell University. Their solution stated exact equations for idealized conditions of heat flow and included a number of assumptions intended to simplify, as far as possible, a complex mathematical calculation. To obtain a comparison between the thermal behavior of a structure under natural weather conditions and the behavior predicted by the Mackey-Wright equations, and to determine the limitations of their assumptions, a research program was initiated at the ASHAE Research Laboratory by the Technical Advisory Committee on Cooling Load□.

The building section chosen was a roof constructed of three homogeneous layers typical of many used today in commercial buildings. Tests were run with the test section exposed to the weather on one side and to a constant temperature space on the other.

This paper presents the results of these tests and compares rates of heat gain and time lags with predicted values based upon the Mackey-Wright equations for multi-layer construction.

### DESCRIPTION OF TEST APPARATUS

The test apparatus was the solar calorimeter used in the research on heat flow through glass and described in Reference 2. In the roof tests, it was not used as a

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□ Exponent numerals refer to References.

□ Personnel: C. O. Mackey, Chairman; R. C. Jordan, Vice Chairman; K. O. Alexander, John Everetts, Jr., H. T. Gilkey, R. H. Heilman, H. W. Heisterkamp, A. J. Hess, A. T. Jörn, C. F. Kayan, J. N. Livermore, P. E. McNall, Jr., I. A. Naman, H. B. Nottage, C. F. Roberts, J. P. Stewart, H. B. Vincent, T. N. Willcox, W. E. Zieber.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

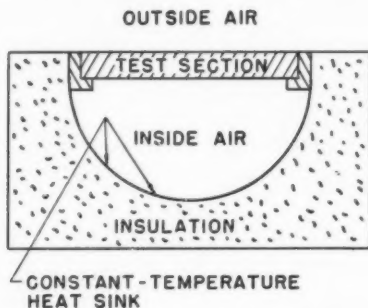


FIG. 1. SCHEMATIC DIAGRAM OF TESTING APPARATUS

calorimeter, but as a constant temperature heat sink. Heat flow at both roof surfaces was measured by heat-flow meters as described later. The heat sink was maintained at a constant temperature by means of a refrigerating unit and thermostatically-controlled electric heaters. Fig. 1 is a schematic sketch of the calorimeter and the test section.

#### CONSTRUCTION AND INSTRUMENTATION OF ROOF SECTION

The test section was approximately 4 ft sq and mounted in a wooden frame that would fit into the opening in the test apparatus. It consisted of a layer of four-ply felt roofing, 2 in. of vegetable-fiber insulation board, and 3 in. of sand-and-limestone-aggregate concrete. As the section was to be tested with the concrete side both in and out, a single ply of felt roofing was cemented on the indoor side to provide surfaces of the same emissivity and solar absorptivity on both sides of the section.

Care was taken in construction of the test section to assure no air blisters between the layers of the roof. The bonding materials used were a quick-setting, rubber-base adhesive and a high-melting-point asphalt.

Temperatures of each surface and each interface of the roof section were measured by sets of 14 copper-constantan thermocouples; 6 were placed diagonally in the center or *test area* of the roof and 8 were placed in the outer or *guard area*. The *guard area* was 8 in. wide, sufficient to insure one-dimensional heat flow through the *test area*. (Temperature readings during the tests proved this to be so.) The surface thermocouples were imbedded slightly in the asphalt surface.

Heat flow was measured by means of four heat-flow meters placed on each surface within the *test area*. These heat-flow meters were the type<sup>3</sup> developed at the ASHAE Research Laboratory. They were bonded to the surfaces of the roof with asphalt and then covered with the same material to provide a surface having the same emissivity and solar absorptivity as the rest of the roof.

Switches were so wired that temperatures across the *test area* could be read individually or in parallel; temperatures in the *guard area* were read individually. The outside heat-flow meters were read individually or in parallel, and the inside heat-flow meters individually or in series. This arrangement made it convenient to check temperatures and heat flows against each other while tests were in progress.

### OTHER INSTRUMENTATION

Solar radiation measurements were made with three Eppley multi-junction pyrheliometers. All were mounted horizontally, two adjacent to the roof and the third on a nearby tower. One pyrheliometer adjacent to the roof measured total radiation and, by means of a shading device, the other measured diffuse radiation. The third could check either of the other two.

A shielded, aspirated thermocouple was used to measure outdoor dry-bulb temperature. Also, at regular intervals, outdoor dry-bulb and wet-bulb temperatures were taken with an aspirated psychrometer.

Low-temperature outdoor radiation measurements were made with a convection-compensated radiometer developed at the ASHAE Laboratory and described in Reference 4.

Wind velocity was measured with a cup-type anemometer and wind direction by a wind vane. General sky conditions were also noted.

The outputs of the thermocouples, heat flow meters, and pyrheliometers were read on an electronic, self-balancing potentiometer. In addition, a record was made on a 16-point high-speed recorder of the following: (a) temperatures of the outside roof surface, (b) temperatures of the inside roof surface, (c) temperatures of the heat-sink surface, (d) heat flowing at the inside and outside surface, (e) amount of total solar radiation, and (f) amount of diffuse solar radiation. The recorder also provided a continuous record of those values most subject to fluctuations, making it possible to average them.

### CALIBRATION OF HEAT FLOW METERS

A number of heat-flow meters were calibrated on a cooled plate by means of the portable calibrator described in Reference 3. From this number, groups of 4 were selected which had calibration constants within 10 percent of each other. The meters were then re-calibrated with the portable calibrator after they were cemented in place on the roof section.

### PHYSICAL PROPERTIES

It was important that the physical values of the materials used in the roof section be accurately ascertained to provide a valid comparison between calculated and observed results. The importance of each value cannot be determined without lengthy and involved calculations. However, particular attention was paid to obtaining accurate conductance values, since they are most subject to variations.

To check the conductances of the various materials, conductance tests were run while the roof was in place in the test apparatus. An aluminum foil container large enough to cover the entire roof surface was put in place and filled with crushed ice. The calorimeter surface (normally the heat sink) was kept at a constant elevated temperature. From the heat flows and the temperatures of the interfaces and surfaces of the test section observed under equilibrium conditions, the overall conductance and the conductance of each layer were computed. These values, except for the concrete, agreed very well with handbook values.

Specific heat values were obtained from handbooks and no experimental determination was made. Since these values are virtually constant for the same type of material, irrespective of other physical properties, further qualification seemed unnecessary.

Densities of the materials were determined from samples obtained at the time of construction of the test sections. Thicknesses of the various layers were measured while the section was being constructed.

The emissivity of the asphalt surfaces was determined by means of a radiometer and apparatus already described<sup>5</sup>. Solar absorptance was computed from spectral light reflectance data<sup>6</sup>. As these data covered approximately 75 percent of the total energy in the solar spectrum, it was assumed that this result could be applied to the entire solar spectrum without much error.

The manner in which the surface coefficients were calculated is shown in Appendix A. All physical properties and surface coefficients are listed in Table 1.

TABLE 1—PHYSICAL PROPERTIES OF MATERIALS USED IN TEST SECTION

MATERIAL	CONDUCTIVITY BTU/(HR) (SQ FT) (DEG F/IN)	SPECIFIC HEAT BTU/(LB) (DEG F)	DENSITY LB/CU FT	THICKNESS IN.
Concrete	6.1	0.22	143.1	3
Insulation	0.42	0.28	21.3	2
Built-up Roofing	1.50	0.25	63.0	$\frac{3}{8}$
Emissivity of Surfaces ( $\epsilon$ )		0.88, no units		
Solar Absorptivity of Surface ( $\alpha$ )		0.91, no units		
Inside Surface Film Coefficient		1.2, Btu/(hr) (sq ft) (deg F)		
Outside Surface Film Coefficient		3.0, Btu/(hr) (sq ft) (deg F)		

#### TESTING PROCEDURE

On the basis of weather forecasts, test periods were so chosen that they might be expected to closely approach the conditions set forth in the Mackey-Wright equations. One of these conditions requires that the steady periodic state be attained. This is reached if the test section is subjected to a sufficient number of identical cycles of sol-air temperature that the temperature of any point in the section goes through the same cycle day after day. Actually, it was found that for this construction, the steady periodic state was attained if the 10-hr period preceding the 24-hr test period was duplicated during the last 10 hr of the test period. As would be expected, because of day-to-day differences in weather, even this condition was not precisely attained and a transient heat-flow component resulted. The effect of this transient component upon the thermal behavior of the roof section was rationally evaluated as described in Appendix B.

All tests were run with the test section in a horizontal position. Readings were taken at regular intervals during the test, for a period of at least 10 hr before the test, and for at least a few hours after the test period. Tests were run with the test section in both the normal (concrete innermost) and reversed positions to study the effect of the order of materials upon the thermal behavior of the roof.

#### OBSERVED RESULTS

Curves of important observed results for Tests Nos. 2 and 3 (the ones chosen for comparison) are plotted in Figs. 2 and 3. Shown are inside and outside surface heat flows and surface temperatures and sol-air temperature. The latter is that

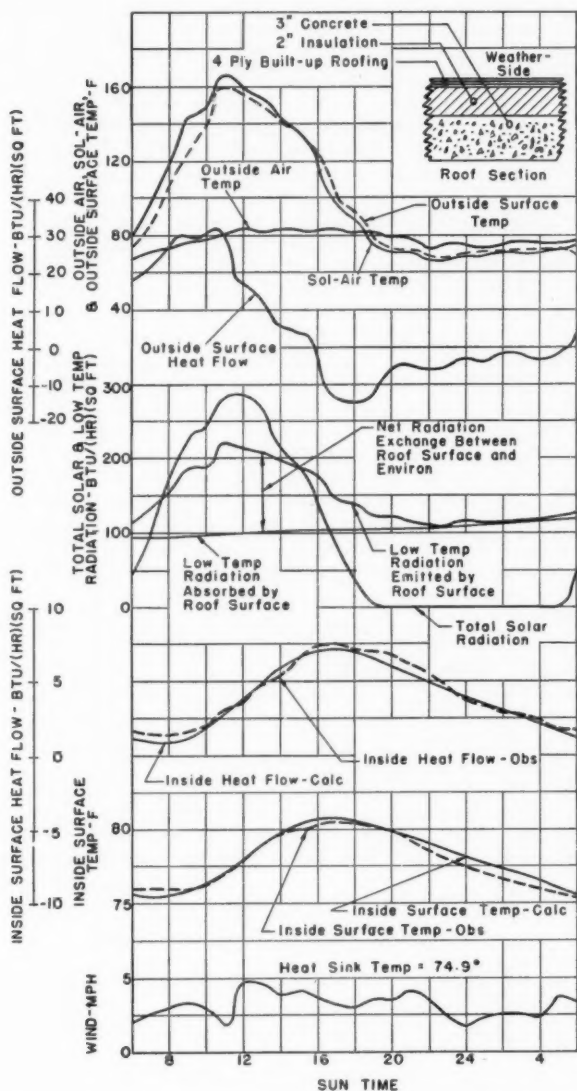


FIG. 2. RESULTS OF TEST NO. 2, DATA OBSERVED JULY 28-29, 1953

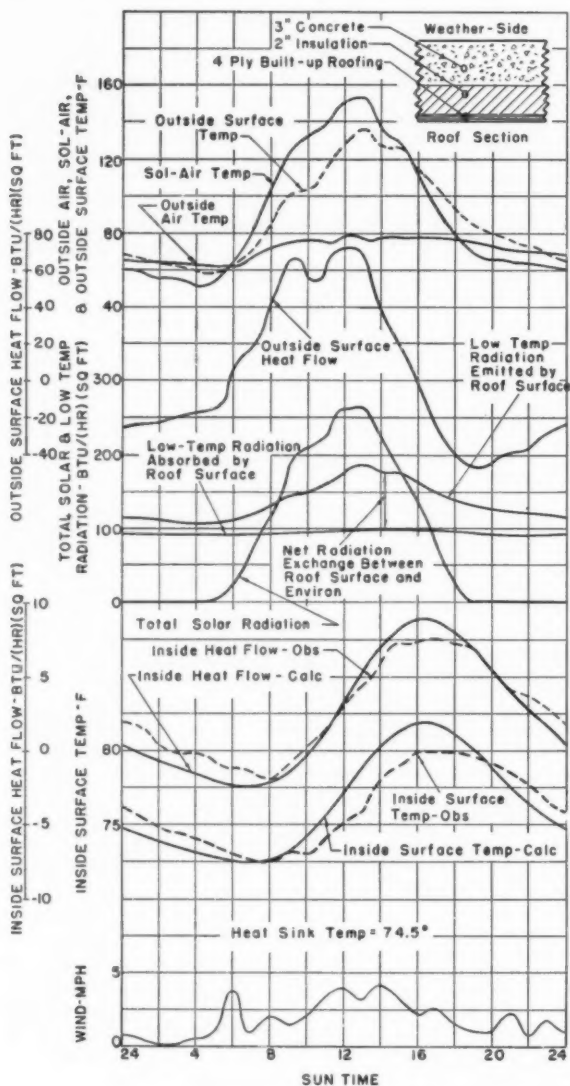


FIG. 3. RESULTS OF TEST NO. 3, DATA OBSERVED AUGUST 11, 1953

equivalent temperature which produces the same rate of heat flow into the weather-side surface of the roof section as the combined effect of outdoor dry-bulb temperature, wind velocity, and radiation incident upon the surface. These observed quantities are also shown in Figs. 2 and 3. Radiation quantities are shown by 2 curves; one is the total solar radiation, which is the sum of the direct and diffuse components; the other is the low-temperature, long-wave length radiation absorbed by the weather-side surface of the roof section. The latter is all the radiation emitted by the outdoor environ exclusive of the sun. The radiation emitted by the roof surface is also shown. The net low-temperature radiation exchange between the roof and its surroundings is the difference between these last 2 curves. It will be noticed that although the low-temperature radiation absorbed by the roof section is a large part of the total incident radiation, the net low-temperature radiation exchange is at all times a negative quantity. The effective radiative temperature of the atmosphere, exclusive of solar radiation, ranged from 30 to 40 deg, well below the air temperature, except during the last part of Test No. 2, when the sky became overcast. This statement perhaps needs some explanation. On a clear, dry day when there is little water vapor or carbon dioxide in the lower atmosphere, a roof exchanges low-temperature radiation principally with the higher, colder atmosphere. As the sky becomes more and more overcast, a larger percentage of the exchange is with the lower air, and when the sky is completely overcast, the effective radiating temperature of the atmosphere is approximately equal to the air temperature. This explanation also clears up a seeming paradox seen in Figs. 2 and 3. During a portion of the night, the air temperature was above the outside roof surface temperature, and yet the roof was giving up heat. The roof was gaining heat by convection, but not as much as it was losing by radiation to the cold upper atmosphere. This difference was balanced by the heat given up by the warmer interior of the roof section.

It can be seen in Figs. 2 and 3 that the sol-air temperature exceeds the outside roof surface temperature while the roof is being warmed up and heat is flowing into the section. At other times the sol-air temperature is lower and drops below the air temperature during portions of the night.

In Table 2 are summarized observed 24-hr heat flows, average temperatures and conductances. The heat flows and the average temperatures are those obtained by integrating the curves of Figs. 2 and 3 and similar curves for Test Nos. 1 and 4. Observed conductance values were obtained from 24-hr average surface and interface temperatures and observed heat flows. The overall coefficient of heat transmission was determined from the average sol-air temperature and the average heat-sink temperature.

#### LAG ANGLES AND DECREMENT FACTORS

The lag angle is defined as the time lag of the peak inside surface temperature behind the peak sol-air temperature and may be expressed in degrees or hours. The decrement factor is a dimensionless ratio which may be defined as follows:

$$\text{decrement factor} = (t_i \text{ max} - t_i \text{ avg}) / (t_e \text{ max} - t_e \text{ avg})$$

where

$t_i \text{ max}$ ,  $t_e \text{ max}$  = maximum inner surface and sol-air temperatures, respectively.

$t_i \text{ avg}$ ,  $t_e \text{ avg}$  = 24-hr average inner surface and sol-air temperature, respectively.

TABLE 2—SUMMARY OF TEST AND CALCULATED VALUES

	UNITS	TEST NUMBER			
		1 <sup>a</sup>	2 <sup>a</sup>	3 <sup>b</sup>	4 <sup>b</sup>
24-hr Heat Flow Rates	Btu/(day)				
Into outside surface — test	(sq ft)	67.4	107.6	45.4	19.7
Stored in roof section — test	"	-26.3	-2.4	-0.6	-24.8
Outside surface-stored — test	"	93.7	110.0	46.0	44.5
From inside surface — test	"	85.4	104.6	61.1	65.9
From inside surface — calc. <sup>c</sup>	"	84.4	99.1	56.4	59.3
24-hr Average Temperatures					
Sol-air — test	F	93.2	100.9	89.5	91.2
Outside surface — test	"	92.3	99.6	88.9	90.4
Inside surface — test	"	77.4	78.0	76.1	76.7
Inside surface — calc. <sup>c</sup>	"	77.5	78.1	76.4	76.4
Heat-sink — test	"	74.8	74.9	74.5	74.7
Conductances	Btu/(hr)				
Surface-to-surface:	(sq ft) (F)				
From 24-hr test	"		0.201	0.198	
Steady-state test	"		0.183	0.183	
Built-up roofing:					
From 24-hr test	"		3.95	3.63	
Steady-state test	"		4.01	4.01	
Insulation:					
From 24-hr test	"		0.236	0.231	
Steady-state test	"		0.211	0.211	
Concrete:					
From 24-hr test	"		2.07	2.31	
Steady-state test	"		2.05	2.05	
Heat transmission coefficient, <i>U</i>					
From 24-hr test	"		0.167	0.169	
From Steady-state conductances	"		0.149	0.149	
Decrement factor					
test	no units	0.0416	0.0379	0.0610	0.0705
calculated <sup>a</sup>	"	0.0429	0.0394	0.0842	0.0893
Lag Angle					
test	degrees	85.	82.	63.	65.
calculated <sup>a</sup>	"	79.	75.	58.	57.
test	hours	5.7	5.5	4.2	4.3
calculated <sup>a</sup>	"	5.3	5.0	3.9	3.8

<sup>a</sup> Concrete on inside of roof section; <sup>b</sup> Concrete on weather-side of roof section; <sup>c</sup> Calculated from Mackey-Wright equations.

This definition is necessary since the sol-air and inside surface temperature curves, as seen in Figs. 2 and 3, are composite and irregular. The observed and calculated lag angles and decrement factors listed in Table 2 were obtained graphically from these curves. If the curves of the sol-air and inside-surface temperatures had been regular sine curves, the decrement factors would have been simply the ratio of their amplitudes.

In Table 3 are listed the harmonic lag angles and harmonic decrement factors as calculated by the Mackey-Wright equations. Values are given for only the first

2 harmonics (*i.e.* fundamental and second harmonic), subsequent harmonics having a negligible effect. The manner in which these values were used to obtain the calculated inside-surface temperature curves is discussed in Appendix A.

### COMPARISON OF RESULTS

From the data taken, 4 usable 24-hr test periods were found, 2 with the concrete innermost and 2 in the reversed position. Fortunately, one test in each position

TABLE 3—CALCULATED HARMONIC LAG ANGLES AND HARMONIC DECREMENT FACTORS

TEST NO.	HARMONIC	LAG ANGLE		DECREMENT FACTOR
		DEGREES	HOURS	
1 & 2	1	90.4	6.0	0.0482
	2	117.4	3.9	0.0290
3 & 4	1	67.0	4.5	0.0923
	2	111.0	3.7	0.0600

(Tests Nos. 2 and 3) was found to closely approximate the required idealized conditions, while in the others some correction for a transient component of heat flow was necessary. It seems wise to confine analysis and comparison principally to Tests Nos. 2 and 3. The conclusions drawn, however, apply qualitatively also to the other tests when corrections for the transient component are made.

In this paper, calculated results refer to values obtained from the Mackey-Wright equations, and observed or test values to those obtained from experimental results. For comparison, the calculated inside surface heat flow and temperature curves are plotted in Figs. 2 and 3. The comparison of calculated and observed inside surface heat flows is of prime importance in this study. To make a comparison of results clear, the assumed conditions used to obtain the calculated results should be pointed out. As stipulated by Mackey and Wright, they are:

1. The temperature of the outdoor air, the total solar radiation, and low-temperature radiation incident upon the outer surface of the roof are cyclic with a period of 24 hr.
2. The temperature of the indoor air (heat sink in these tests) is constant.
3. The rate of heat transfer by convection and radiation between the outdoor air and the outside surface per degree temperature difference is constant.
4. The rate of heat transfer by convection and radiation between the indoor surface and the indoor air (heat sink) per degree temperature difference is constant.
5. The roof is composed of three layers arranged in series in the direction of heat flow, each layer being itself a single homogeneous material.
6. The three layers make perfect thermal contact at their adjoining faces.

Primarily, only conditions 3 and 4 need be considered in the analysis of results. Condition 1 was either closely approximated or rationally corrected for, condition 2 was maintained during the testing periods, and conditions 5 and 6 were met when the test section was constructed.

In Test No. 2 (concrete innermost) there is excellent agreement between the calculated and observed inside surface heat flows (Fig. 2). This means that the lag angles and decrement factors obtained by the two methods agree closely. It

will be noticed in Fig. 2 that the inside surface heat flow is positive (from the inside surface to the heat sink) at all times. Any effect caused by an assumption of a constant inside-surface coefficient is negligible since, within the range of temperatures encountered in these tests, the coefficient is virtually constant as long as the direction of heat flow remains the same. The small discrepancies in the heat flows shown in Fig. 2 can be attributed to small changes in the heat-sink temperature.

In Test No. 3, agreement between calculated and observed inside surface heat flows is not quite as good. The lag angles agree and, although the summation of observed and calculated heat flows for a 24-hr period agree closely, the decrement factor of the observed curve is less than that of the calculated curve. Major cause of discrepancies, it seems, lies in the assumption of a constant inside-surface coefficient. In Test No. 2 with constant direction of heat flow, this assumption was valid, but not in Test No. 3. In this test there was a reversal of heat flow and subsequently a change in surface coefficient. The radiation portion of the coefficient remains the same irrespective of the direction of heat flow, but the convection portion is much higher for upward heat flow. Proper evaluation of the effect cannot be readily computed. The use of an analogue computer or similar device capable of having a step change made in the coefficient would be necessary.

The effect of order of materials is clearly shown in the observed results. As predicted by the Mackey-Wright equation, a roof with a material of lesser density outermost has a greater lag angle and a smaller decrement factor than for the reversed position. It is also seen in Table 2 that for all tests there is reasonably good agreement between the summation of the outside surface heat flow minus the heat storage, and the calculated and observed inside surface heat flows. The heat storage was evaluated using the average surface temperatures of each layer and assuming a straight-line relationship of temperature with distance from the surface. However, the temperature gradient through a particular layer is not necessarily a straight-line function and consequently, the heat storage is an approximation. This could account for a portion of any differences present.

Good agreement is also shown between conductance values obtained by steady-state conductance tests and observed values (both previously described). Conductance values were not calculated for Test Nos. 1 and 4, since appreciable heat storage took place and proper evaluation of its effect was not possible.

### CONCLUSIONS

1. The calculated inside surface heat flow obtained using Mackey and Wright's equations agree closely with observed values, better agreement being achieved when the inside surface heat flow remains in one direction, than when a reversal takes place.

2. The predicted effect of order of materials was substantiated. With the material of higher density outermost, the decrement factor is larger and the lag angle smaller than for the reversed position. Practically speaking, this means that the maximum rate of heat flow from the inside surface is larger and the time lag smaller when the material of higher density is outermost.

3. The limitations of assuming a constant outside surface coefficient could not be determined. A good recommendation seems to be to use the average daytime wind velocities for evaluating the outside surface coefficient. This is discussed in Appendix A.

4. The assumption of a constant inside surface coefficient is valid only when heat flow is in one direction. For horizontal surfaces, reversal of heat flow causes a significant change in the coefficient with a subsequent change in heat-flow.

It should be pointed out that the tests are for only 2 particular roof constructions (actually 1 roof in 2 positions). For other types of construction having different thermal properties and orientation, observed results will be different. In the extreme case of a horizontal roof having a very small thermal resistance and a reversal of inside surface heat flow, the comparison of calculated and observed results may also differ. In this case the inside surface film coefficient would be a bigger part of the overall thermal resistance and would probably have a greater effect.

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5. ASHVE RESEARCH REPORT NO. 1528—Measurement of Angular Emissivity, by Aydin Umur, G. V. Parmelee and L. F. Schutrum (ASHVE TRANSACTIONS, Vol. 61, 1955, p. 111).
6. Private communication from Dr. W. J. Arner (Libby-Owens-Ford Glass Co.; July, 1953).

#### APPENDIX A

##### Calculation Methods

##### Surface Coefficients

To calculate heat flow rates through a building section in the steady periodic state, it is necessary to know the physical properties of the section, the indoor and outdoor surface coefficients and the sol-air temperatures. The physical properties were determined experimentally as described earlier. The surface coefficients were calculated from a heat balance on the outer roof surface. The heat balance equation is:

$$\alpha I_T = \epsilon(\sigma T_s^4 - R) + h_o(t_o - t_a) + q_{in} \dots \dots (A-1)$$

where

- $\alpha$  = solar absorptivity of roof surface, no units.
- $I_T$  = total incident solar radiation, Btu per (hour) (square foot).
- $\epsilon$  = emissivity of roof surface, no units.
- $\sigma$  = Stefan-Boltzmann constant =  $0.173 \times 10^{-8}$  Btu per (hour) (square foot) (Fahrenheit degree absolute)<sup>4</sup>.
- $R$  = incident low-temperature radiation, Btu per (hour) (square foot).
- $t_o, T_o$  = outside roof surface temperature, Fahrenheit and Fahrenheit absolute, respectively.
- $t_a, T_a$  = outdoor air temperature, Fahrenheit and Fahrenheit absolute, respectively.
- $q_{in}$  = measured heat flow into outer roof surface, Btu per (hour) (square foot).
- $h_o$  = convective conductance, Btu per (hour) (square foot) (Fahrenheit degree).

The methods by which  $\alpha$  and  $\epsilon$  were obtained have been described in the paper.  $I_T$ ,  $R$ ,  $t_a$ ,  $t_s$  and  $q_{in}$  were determined from test data. Thus, with all other values known in Equation A-1,  $h_o$  could be computed.

In calculating the sol-air temperatures it is necessary to consider the low-temperature radiation exchange. This can be facilitated by rewriting Equation A-1 as follows:

$$\alpha I_T = \epsilon \sigma (T_s^4 - T_a^4) + \epsilon (\sigma T_a^4 - R) + h_o(t_s - t_a) + q_{in} \dots (A-2)$$

A radiation exchange coefficient,  $h_r$  can now be defined by the relationship:

$$h_r(t_s - t_a) = \epsilon \sigma (T_s^4 - T_a^4) \dots (A-3)$$

This value of  $h_r$  can be added to  $h_o$  to give a combined outside surface coefficient,  $h_o + h_r$ . Values of  $h_o$  and  $h_r$  computed from the test data for all four tests varied from 1.14 to 3.82 Btu/(hr) (sq ft) (F deg).

Because calculations by the Mackey-Wright equations required a constant surface coefficient, an average value was computed by weighing  $h_o + h_r$  according to  $(t_s - t_a)$ . This yielded an average substantially equal to 3.00 Btu/(hr) (sq ft) (F deg).

The inside surface coefficient was obtained by dividing hourly measured values of inside surface heat flow by the observed temperature difference between the inside roof surface and the heat sink. These values were then averaged as before. The range of values varied from 0.94 to 1.53 Btu/(hr) (sq ft) (F deg), the larger values occurring when the heat flow was upwards. The average value was 1.20 Btu/(hr) (sq ft) (F deg).

#### Sol-Air Temperatures

The sol-air temperature is that equivalent temperature,  $t_e$ , which combines radiation, wind, and air temperature in such a way as to give the same rate of heat flow into the weather surface of the roof as actually takes place, that is,

$$q_{in} = (h_o + h_r)(t_e - t_a) \dots (A-4)$$

from which

$$t_e = t_a + \frac{q_{in}}{h_o + h_r} \dots (A-5)$$

The sol-air temperature can also be computed from Equation A-2 thus:

$$t_e = \frac{\alpha I_T - \epsilon (\sigma T_a^4 - R)}{h_o + h_r} + t_a \dots (A-6)$$

Because  $q_{in}$  and  $t_a$  were observed in these tests, for the sake of accuracy,  $t_e$  was evaluated by Equation A-5. If all values in Equation A-6 are known accurately, the same results would be obtained. Comparison of the sol-air temperatures obtained by the two methods showed very close agreement.

In the practical problem of load estimating, Equation A-6 must be used. Although  $t_a$  is not known, in such cases a rough estimate is sufficiently accurate for determining  $h_r$ , because it is a function of the mean of  $t_s$  and  $t_a$ , the latter is known, and  $h_r$  does not change too rapidly with mean temperature. A further practical observation is that these tests showed that a simple average of the daytime values of  $h_o + h_r$  is a good approximation to use in practical calculations where  $h_o + h_r$  is known to vary.

#### Harmonic Analysis

For computation purposes, the sol-air temperature must be expressed as a mathematical function. A Fourier analysis was made of the curve by the method described

by Alford, et al†. This yields an equation as follows:

$$t_o^* = t_{eo} + a_1 \cos \omega\theta + b_1 \sin \omega\theta + a_2 \cos 2\omega\theta + b_2 \sin 2\omega\theta + \dots \quad (\text{A-7})$$

where

- $t_{eo}$  = 24-hr average sol-air temperature, Fahrenheit.  
 $a, b$  = Fourier or harmonic coefficients; subscripts denote the harmonic number, no units  
 $\theta$  = Time, hours  
 $\omega$  = Degrees per hour

This equation can be reduced to a more usable form, thus:

$$t_o^* = t_{eo} + t_{e1} \cos (\omega\theta - \beta_1) + t_{e2} \cos (2\omega\theta - \beta_2) + \dots \quad (\text{A-8})$$

where

$$t_{e1} = \sqrt{a_1^2 + b_1^2}; t_{e2} = \sqrt{a_2^2 + b_2^2}; \text{ etc.} \\ \tan \beta_1 = b_1/a_1; \tan \beta_2 = b_2/a_2; \text{ etc.}$$

### Computation of Inside Surface Temperatures and Heat Flows

The inside-surface temperature,  $t_i$ , is computed from the following equation

$$t_i^* = t_{io} + M_{L1} t_{e1} \cos (\omega\theta - \beta_1 - m_{L1}) + M_{L2} t_{e2} \cos (2\omega\theta - \beta_2 - m_{L2}) + \dots \quad (\text{A-9})$$

where

- $M_{L1}, M_{L2}$  = amplitude decrement factors, no units.  
 $m_{L1}, m_{L2}$  = phase lag angle, degrees.

The amplitude decrement factors and the phase lag angles are computed by the Mackey-Wright equations.  $t_{io}$  is the 24-hr average inside-surface temperature calculated from the relationship:

$$U(t_{eo} - t_e) = h_i(t_{io} - t_e) \dots \quad (\text{A-10})$$

where

$t_e$  = constant heat sink temperature, Fahrenheit.

The  $U$  value was determined from the relationship

$$\frac{1}{U} = \frac{1}{h_o + h_r} + \frac{1}{C} + \frac{1}{h_i} \dots \quad (\text{A-11})$$

where

$C$  is the steady-state surface-to-surface conductance value and  $h_o + h_r$  and  $h_i$  are the surface coefficients determined as described above. The heat flow rate,  $q$ , from the inside surface of the roof section to the heat sink is

$$q = h_i (t_i - t_e), \text{ Btu/(hr) (sq ft)} \dots \quad (\text{A-12})$$

## APPENDIX B

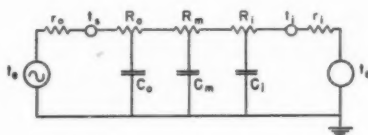
### Evaluation of Transient Heat Flow Component

Because of the presence of a transient component of heat flow, a study was made to estimate its magnitude and duration in all tests. A thermal circuit to represent the

† Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan, and F. O. Urban (ASHVE TRANSACTIONS, Vol. 45, 1939, Appendix III, p. 393).

\* In Fahrenheit degrees.

roof section was drawn. This consisted of a number of resistances and capacitances, and a cyclic input arranged as in Fig. B-1. The input function was a Fourier representation of the sol-air temperature. The solution of any problem concerning this circuit is similar to that for an analogous electrical circuit. Solution of the three simultaneous first-order differential equations which represented this circuit, showed that the transient component was substantially zero at the end of 10 hr. About the same result was obtained from a solution for a circuit using 2 capacitances instead of 3 and



$R_o, R_m, R_i$  = thermal resistances of the outer, middle, and inner layers, respectively, Btu per (hour) (square foot) (Fahrenheit degree).

$C_o, C_m, C_i$  = thermal capacitances of the outer, middle, and inner layers, respectively, Btu per (square foot) (Fahrenheit degree), (connected to center of resistance).

$r_o, r_i$  = thermal resistances of the outer and inner surface films, respectively, Btu per (hour) (square foot) (Fahrenheit degree).

$t_e$  = cyclic sol-air temperature, Fahrenheit.

$t_o$  = constant heat-sink temperature, Fahrenheit.

$t_s, t_i$  = outside and inside surface temperatures, respectively, Fahrenheit.

FIG. B-1. THERMAL CIRCUIT OF TEST SECTION

solved on an electronic differential analyzer. Because the time lag of the roof section was 5 to 6 hours, it was reasoned that the contribution of the transient component to the indoor surface of the roof section could be obtained from experimental observations as follows: The surface temperatures observed during the first 5 or 6 hours immediately following the test period are steady-periodic-state values resulting from the sol-air temperature of that period. Because of the time lag, the sol-air temperature of the second period has as yet, had no effect. Furthermore, the transient heat flow component of the first period died out 14 hours earlier. Therefore, the transient contribution is the difference between the temperatures observed during the first 5 or 6 hours of the period being analyzed and the temperatures of the corresponding hours of the second cycle or period. A plot of these temperature differences against time and extrapolation to 0 at 10 hours, therefore, should yield the complete transient contribution. The heat flows computed from this temperature difference curve checked roughly the heat stored in the roof section found by computation from the initial and final temperatures observed during the period of analysis. Corrections to the observed indoor surface temperatures were therefore made according to this principle.

## DISCUSSION

C. O. MACKEY, Ithaca, New York (WRITTEN): This report of experimental studies of heat flow through composite roofs is a valuable addition to the literature on heat transfer in the periodic state. It should be emphasized that the measured rates of heat transfer from the room-side surface of the composite roof are instantaneous rates of

heat transfer to uniform surroundings (air and surface) at the same constant temperature. These are not instantaneous cooling loads on cooling systems of conventional design; only the convection heat transfer, which amounts to about 30 percent of the total in these studies, is an instantaneous contribution to the cooling load; the radiation emitted by the indoor surface must be absorbed and transferred from the surrounding surfaces to the air by convection before it is felt as cooling load.

The purpose of these experiments was to determine whether the available analytical solutions give reasonable results for the instantaneous rates of heat transfer from the room-side surfaces of composite constructions. The answer seems to be that they do if the two surface coefficients of heat transfer are carefully selected.

In particular, much care seems to be necessary in selecting the room-side surface coefficient for combined heat transfer by radiation and convection. In some cases, the room-side surface of the roof will exchange radiant energy with surfaces that are *not* at the air temperature. This will modify the value that should be chosen for the *combined* surface coefficient, when this coefficient is defined as the rate of heat transfer per degree of temperature difference between room-side surface and *air*.

The value of this paper would be increased by giving the temperature coefficients and lag angles actually used by the authors in the Fourier series of Equation A-8 for the four tests. It is hoped that the authors will supply this information in their closure.

Appendix A is more important than its position in this paper would seem to indicate. This appendix gives a brief explanation of the present concept of sol-air temperature. This concept has been extremely useful to those concerned with periodic heat flow.

Appendix B presents an original and interesting method of correcting the *observed* room-side surface temperature to obtain *calculated* values of these temperatures for true periodic conditions when the weather-side conditions are not exactly periodic. It represents a contribution to the technique of the experimental study of periodic heat flow.

HARRY BUCHBERG\*, Los Angeles, Calif.: The authors should be commended for a carefully executed experimental program which adds to the knowledge of periodic heat flow through specific materials under particular boundary conditions. The big problem is how to apply this data, determined for single structural sections, when dealing with an entire structure and different boundary conditions.

Examination of the thermal circuit representing a single or multi-room structure will show that generally the complex impedances representing the parallel conduction paths are dependent not only on the thermal properties of the single structural component but also on what is happening at other parts of the circuit. As presented in this paper, the decrement factor and lag angle expresses some average value of the complex impedance to heat transfer through a single structural component for a particular sol-air temperature cycle and set of boundary resistances. The use of these quantities is further restricted to the case where all inside surface temperatures are uniform at all times during the cycle and where there are no heat sources feeding into the inside surfaces.

It appears that a better approach to the problem for the case where the inside air temperature is constant might be to determine values of complex impedances which characterize a particular structural component independent of the sol-air temperature and boundary resistances. These complex impedances could easily be determined using a network analogue computer for a frequency corresponding to the fundamental sinusoid and for additional harmonics if necessary. With these values of impedance, instantaneous heat rates into the space could be determined for any sol-air temperature cycle and any boundary resistances.

AUTHORS' CLOSURE (Mr. Vild): On behalf of the co-authors and myself, I would like to thank Professor Mackey and Mr. Buchberg for their comments on this paper.

\* Department of Engineering, University of California at Los Angeles.

We are in perfect agreement with Professor Mackey's comment that the choice of film coefficients is very important in the calculated results, particularly at the inside surface. In our calculations, we chose coefficients that were temperature-difference weighted averages of coefficients obtained from test results.

The following are the temperature coefficients and lag angles used in equation A-8. These are obtained from a Fourier analysis of the sol-air temperature as described in Appendix A.

TEST NO.	$h_{eo}$	$h_{ei}$	$h_{et}$	$\beta_i$	$\beta_t$
1	93.2	53.9	16.1	0.3 deg	12.3 deg
2	100.9	44.9	17.4	0.0 deg	-1.5 deg
3	89.5	48.5	17.8	7.0 deg	-0.3 deg
4	91.2	53.2	18.5	5.9 deg	4.5 deg

Mr. Buchberg has outlined a procedure for applying thermal-circuit techniques to the solution of a practical cooling-load problem. The complexity of boundary conditions for a practical case as compared with the simplified case used in our program are realized. Our objectives in this program were "to obtain a comparison between the thermal behavior of a structure under natural weather conditions and the behavior predicted by the Mackey-Wright equations, and to determine the limitations of their assumptions". This limits our choice of "indoor" conditions. Mr. Buchberg also suggests an alternate manner of solution which appears to be an extremely complex analytical approach.



**1546**

## VENTILATION OF COMMERCIAL LAUNDRIES

By SIDNEY MARLOW\*, NEW YORK, N. Y.

THE COMMERCIAL laundry has always been confronted with the problem of heat and moisture removal from the workroom. Ventilation must be provided to keep temperature and humidity within limits considered safe for workers. The main pieces of laundry equipment which subject the workroom environment to high temperature and humidity are the flatwork ironers, washers and presses.

### FLATWORK IRONERS

A canopy hood located over the rolls has been found to be the most practicable means for ventilating flatwork ironers. Several such hoods are commercially available. They usually have hinged side panels with closely spaced glass panes (Fig. 1). Hinged panels make possible easy access to the ironer for repadding rolls, inspection and maintenance. Where feasible, the bottom edge of the hood should be placed flush with the top of the sides of the ironer. In this way, air need be exhausted only through the working openings of the ends of the ironer.

Two factors must be considered in determining the rate of ventilation for canopy exhaust. First, a minimum control velocity must be maintained at all the openings between the bottom edge of the hood and the top edge of the machine. This velocity is essentially the minimum capture velocity for steam vapor and may be fixed as 50 fpm. Secondly, we may safely assume that the hood and duct will approach the temperature of the exhausted air passing through it. The temperature of the exhausted air should, therefore, be limited to a temperature of 125 F to minimize heat transfer back to the workroom, as well as to make the hood safe to touch.

By means of heat balances, it is possible to estimate the exhaust volume necessary to meet the 2 given requirements. In determining the heat load which must be dissipated, only the sensible portion of the delivered boiler horsepower must be considered. From Table 1, it will be seen that about 65 percent of the rated delivered boiler horsepower is sensible heat.

$$Q = 794 \text{ (bhp)} \dots \dots \dots (1)$$

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\* Industrial Hygiene Engineer, Division of Industrial Hygiene, New York State Department of Labor. Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

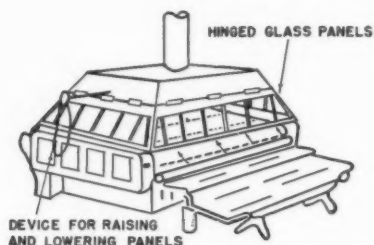


FIG. 1. CANOPY HOOD FOR FLATWORK IRONER

giving the ventilation rate  $Q$  in cubic feet per minute as a function of the rated flatwork ironer boiler horsepower (bhp) is derived in the Appendix. The most common flatwork ironers and the ventilation rates as calculated from Equation 1 are shown in Table 1, Column 5.

TABLE 1—SENSIBLE HEAT LOADS AND MINIMUM VENTILATION RATES FOR FLATWORK IRONERS

COLUMN No.	1	2	3	4	5
IRONER INCHES	RATED <sup>a</sup> BOILER HP BHP	APPROX <sup>b</sup> PRODUCTION LB/HR	LATENT BHP	PERCENT RATED BHP SENSIBLE	MINIMUM <sup>c</sup> VENTILATION RATE CFM
2-Roll-100	3.1	75-85	1.2	66	2500
2-Roll-120	3.8				3000
4-Roll-100	6.3	150-165	2.3	67	5000
4-Roll-110	6.8				5400
4-Roll-120	7.6				6000
6-Roll-100	9.0	225-250	3.5	65	7100
6-Roll-110	9.9				7900
6-Roll-120	10.7				8500
8-Roll-100	12.8	300-330	4.6	66	10,000
8-Roll-120	14.2				11,000
12-Roll-120					15,000

<sup>a</sup> Determination of Laundry Power Plant Requirements, Special Report No. 146. Bhp ratings taken from Table 4, p. 16 (*American Institute of Laundering*, Joliet, Ill.).

<sup>b</sup> Wearing apparel and flat work

<sup>c</sup> Calculated by Formula —  $Cfm = 794 (bhp) = 794 \times \text{Column 1}$ .

In determining the ventilation rate it is necessary to compare the ventilation rate based upon 50 cfm per square foot of net opening with that shown in Table 1 and then to select the greater of the two. For example, canopy hoods similar to that shown in Fig. 1 are to be used for a 2-roll-120 in. and an 8-roll-120 in. flatwork ironer. The sides of the canopy hood are flush with the top side pieces of the ironer leaving, we may assume, net working openings at each end  $12 \times 1$  ft, or a total of 24 sq ft for both sides. From Table 1, the ventilation rates are 3000

and 11,000 cfm respectively for the 2-roll-120 in. and 8-roll-120 in. ironers. Since these rates exceed the 1200 cfm required to maintain the minimum 50 fpm control velocity, the ventilation rates from Table 1 are satisfactory. On the other hand, if the net working openings were 12 X 3 ft, a ventilation rate of 3600 cfm for the 2-roll ironer would be required to maintain the minimum 50 fpm control velocity.

In addition to ventilating the ironer, the hood also provides protection for the padded rolls and chests from dust and lint and is a safety guard.

### WASHERS

The problem of determining the amount of ventilation required for workrooms where washers are located is complicated by the lack of available data involving heat losses through the washer shells. In addition, the washers are operated through a definite cycle where the water changes are rather frequent. The washer water temperatures will vary from cold to hot, the exact temperatures depending upon the washing formula being used. The maximum temperature encountered in practically all washers is 180 F and is used only during the rinsing operations. Furthermore, the effective area of the washer shell through which the heat is

TABLE 2—GENERAL VENTILATION RATES FOR METAL WASHERS

NOMINAL SIZE INCHES	VENTILATION RATE, CFM
24 x 24	340
24 x 36	450
30 x 30	530
30 x 36	600
30 x 48	740
36 x 36	760
36 x 54	1000
36 x 64	1200
42 x 36	940
42 x 54	1200
42 x 64	1400
42 x 72	1500
42 x 84	1700
42 x 96	1900
44 x 36	1000
44 x 54	1300
44 x 60	1400
44 x 72	1600
44 x 84	1800
44 x 96	2000
44 x 108	2200
44 x 120	2400
54 x 84	2300
54 x 96	2600
54 x 108	2900
54 x 120	3100

Note: Use 25 percent of the cfm rates for corresponding wood washers.

transmitted to the surrounding air is practically indeterminate. All these factors make it difficult to obtain ventilation rates with any degree of accuracy.

At best, ventilation rates may be estimated by assuming an average surface shell temperature of 160 F and an overall heat transfer coefficient of 3.0 Btu per (hr) (sq ft) (F deg) for a metal shell. The effective area of heat transfer may be considered to be the surface of the cylinder. Washers are nominally specified in terms of the cylinder diameter and length; the cylinder surface area is thus easily obtainable.

The expression derived in the Appendix gives the ventilation rate  $Q$  in cubic feet per minute as a function of the nominal length  $N$  and diameter  $d$  (both in inches) of a metal washer as

$$Q = 0.39d [N + (d/2)] \quad \dots \dots \dots (2)$$

A list of the various size washers with the exhaust volumes as calculated from Equation 2 is given in Table 2.

The exhaust rates for washers in Table 2 have been based upon maintenance of a maximum dry-bulb temperature of 100 F in the workroom. In many plants

#### NOMENCLATURE

- $A_s$  = effective surface heat transfer area of washer, square feet.
- $B$  = conversion factor, 33,500 Btu per hour (boiler horsepower).
- bhp = rate of heat transfer from ironer and press units, boiler horsepower.
- $c_p$  = heat capacity of water vapor = 0.48 Btu per (pound) (Fahrenheit degree).
- $D$  = dry wash entering and leaving system, pounds per hour.
- $d$  = nominal diameter of cylinder of washers, inches.
- $G$  = dry air entering and leaving system, pounds per hour.
- $H_1$  = absolute humidity of air entering system, pounds moisture per pound dry air
- $H_2$  = absolute humidity of air leaving system, pounds moisture per pound dry air
- $\Delta h$  = change in specific enthalpy of dry air from  $t_1$  to  $t_2$ , Btu per pound dry air.
- $L$  = latent heat of vaporization of moisture in clothing being pressed, Btu per pound moisture.
- $M$  = moisture content of clothing entering system, pounds moisture per pound dry wash.
- $N$  = nominal length of cylinder of washers, inches.
- $Q$  = ventilation rate, cubic feet per minute.
- $q$  = total heat transferred from presses comprising unit, Btu per hour.
- $s_a$  = average humid heat capacity of air between 100 F and 125 F = 0.30 Btu per pound humid air (F deg).
- $s_1$  = humid heat capacity of outdoor air entering workroom = 0.245 Btu per pound dry air (F deg) at  $t_1 = 90$  F and  $t_w = 75$  F.
- $s_f$  = heat capacity of cotton flatwork = 0.32 Btu per (lb) (F deg).
- $T$  = wet-bulb temperature of air in workroom, Fahrenheit degrees.
- $t_1$  = temperature of flatwork entering ironer, Fahrenheit degrees.
- $t_0$  = temperature of flatwork leaving ironer = 150 F.
- $t_1$  = dry-bulb temperature of air entering system, Fahrenheit degrees.
- $t_2$  = dry-bulb temperature of air leaving system, Fahrenheit degrees.
- $t_w$  = wet-bulb temperature of air entering system, Fahrenheit degrees.
- $\Delta t_1$  = maximum temperature rise of canopy exhaust air from flatwork ironer, Fahrenheit degrees.
- $\Delta t_2$  = difference in temperature between shell of washer and surrounding air, Fahrenheit degrees.
- $U$  = washer overall heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
- $v$  = specific volume of air (at 70 F,  $v = 13.4$  cu ft per lb).

the spent wash water is dumped directly on the floor and is disposed of through floor drains. On the floor the water presents a large water to air interface; this probably makes the humidity rather than the dry-bulb temperature the controlling factor in determining the amount of ventilation. Since there is no means of estimating the amount of moisture being transferred to the air, it is suggested that the exhaust volumes in Table 2 be multiplied by a factor of 2 in determining the ventilation rate of washrooms where such practices exist.

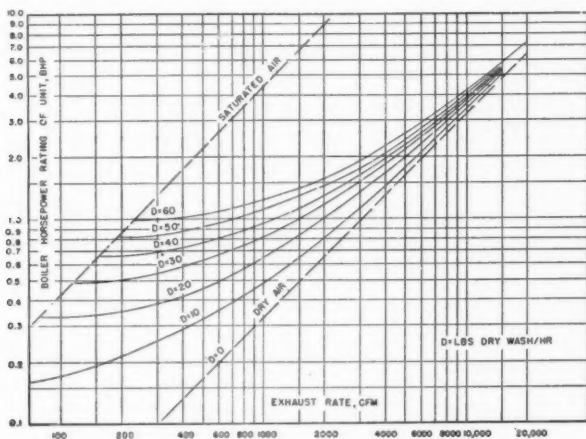
For corresponding size wood washers, the washer overall heat transfer coefficient may be assumed to be one-fourth the value for metal washers. Therefore, for wood washers of the same size as indicated in Table 2, 25 percent of the exhaust volume listed in Table 2 should be used.

### PRESSES

The use of local exhaust ventilation as a means of heat and moisture removal from shirt and garment presses is not practicable because of the nature of the pressing operation. A method of general room ventilation of the press area is therefore preferred.

$$\text{bhp} = (60/B) (Q/V) s_1 (t_2 - t_1) + (MD/B) [L + c_p (t_2 - T)]. \quad (3)$$

is the expression derived in the Appendix and in terms shown under Nomenclature, giving the ventilation rate,  $Q$ , for presses as an implicit function of bhp. Since a solution of Equation 3 involves trial and error calculations, it is simpler to solve for  $Q$  graphically.



Air entering workroom assumed to be at a dry-bulb temperature of 90 F and a wet-bulb temperature of 75 F. Dry-bulb temperature limited to 100 F; 10 F rise in dry-bulb temperature

FIG. 2. EXHAUST RATE VS. RATED BOILER HORSEPOWER OF PRESS UNIT

Fig. 2 represents a plot of Equation 3 showing the required exhaust volume  $Q$  as a function of the maximum boiler horsepower of a unit handling an average of  $D$  pounds of dry clothing per hour. The straight line labeled *Saturated air* represents moisture saturation of the air at the maximum production rate  $D$ .

A survey of several of the large New York metropolitan commercial laundries revealed that the representative production rate for shirt pressing units may be taken as 12 lb of dry wash per operator per hour. However, 2 commercial units, one of which incorporated a double bosom press, were able to produce as much as 15 lb per operator per hour. It was also found that a production rate of 12 lb per operator per hour is representative for general apparel pressing units where 1 operator is assigned to each general apparel unit.

The determination of the maximum boiler horsepower consumed by the various press units would require calorimetric measurements of the steam at the press inlet and outlet as well as weighing the condensate over a definite period of time. For the same type of unit this value would, in all probability, vary from plant to plant, depending upon the condition of the steam being fed to the press, the nature of production, and the design and type of bucks used in the presses. For the purpose of assigning a definite maximum boiler horsepower rating to the various type presses commonly used, the manufacturers rated boiler horsepower of the various presses were tabulated and a representative rating selected for each type press as shown in Tables 3 and 4.

A list of practically all the various types of units used with recommended exhaust rates is given in Tables 5 and 6. Since one operator is assigned to an ap-

TABLE 3—BOILER HORSEPOWER (BHP) RATINGS FOR SHIRT UNIT PRESSES

SHIRT UNIT PRESSES	ASSIGNED BHP	APPROXIMATE SIZE OF BUCK INCHES
Body Press.....	1.0	35 x 20 x 12
Bosom Press.....	1.5	40 x 18 x 11
Combination Bosom and Body Press.....	1.5	40 x 18 x 11
Collar and Cuff Press.....	0.75	20 x 6, 10 x 7
Cuff and Gusset Press (Double Cuff Press).....	0.75	8 x 8 x 4
Double Collar Press.....	0.75	20 x 8 each
Double Sleeve Press.....	0.75	23 x 11
Double Sleeve Form.....	0.25	.....
One Lay Double Sleeve Press.....	0.54	24
Single Collar Press.....	0.30	20 x 7
Two Lay Sleeve Press.....	0.27	24
Yoke Press.....	0.32	20 x 11

TABLE 4—BOILER HORSEPOWER (BHP) RATINGS FOR APPAREL PRESSES

APPAREL PRESSES	ASSIGNED BHP	APPROXIMATE SIZE OF BUCK, INCHES
Coverall Body Press.....	0.75	33 x 16 x 13
Mushroom Press.....	0.37	18 x 10
Mushroom Press, Tapered.....	0.47	21 x 12
Apparel Press, Rectangular.....	1.9	53 x 18
Apparel Press, Tapered.....	1.9	50 x 17 x 9

TABLE 5—GENERAL VENTILATION RATES FOR SHIRT PRESSING UNITS

PRESSES COMPRISING UNIT	OPERATORS	PRODUCTION LB/HR	MAXIMUM UNIT, BHP	EXHAUST RATE CFM
1. Collar & Cuff Combination-Bosom and Body Two Lay Sleeve	2	24	2.52	6200
2. Collar & Cuff Yoke Bosom Double Sleeve Form	3	36	2.82	6600
3. Collar & Cuff Yoke Bosom Two Lay Sleeve	3	36	2.84	6600
4. Collar & Cuff Yoke Combination-Bosom and Body Two Lay Sleeve	2	24	2.84	7000
5. Collar & Cuff Bosom Body Double Sleeve Form	3	24	3.50	9000
6. Double Cuff & Gusset Single Collar Yoke Combination-Bosom and Body Double Sleeve Double Sleeve Form	4	60 <sup>a</sup>	3.87	9000
7. Two Lay Sleeve (2) Collar and Cuff Yoke Body Bosom	4	48	4.11	10,000
8. Double Sleeve Yoke Combination-Bosom and Body Body Single Collar Double Cuff Double Sleeve Form	4	48	4.12	10,000
9. Yoke Body Bosom Collar and Cuff One Lay Double Sleeve	3	36	4.11	11,000
10. Two Lay Sleeve (2) Collar and Cuff Yoke Body Bosom	3	36	4.11	11,000
11. Combination-Bosom & Body Body Double Cuff Collar & Cuff Yoke Double Sleeve Form	4	48	4.57	12,000
12. Two Lay Sleeve (2) Double Collar Body Bosom Yoke Double Cuff	4	48	4.86	12,500
13. Two Lay Sleeve (2) Combination-Bosom and Body Double Cuff Double Sleeve Yoke Body	4	48	4.86	12,500
14. Double Sleeve Double Cuff & Gusset Single Collar Body Double Sleeve Form Double Bosom (Tiltor)	4	60 <sup>a</sup>	6.05	16,000

<sup>a</sup> High production, 15 lb/operator per hr. Average production, 12 lb/operator per hr.

parel unit, the average production rate of 12 lb per operator per hour applies for the unit. For combinations of presses comprising units not listed, the exhaust volume may be obtained by determining the maximum unit boiler horsepower and production rate and using Fig. 2.

In order to increase the productive capacity of flatwork ironers, some laundries *precondition* the extracted wash before sending it to the ironers. The extracted wash is dried in a heated tumbler ventilated to the outside usually to a moisture content less than 0.1 lb of water per pound of dry wash. With such a low moisture content, the wash can be passed through the rolls at a much faster rate increasing the productive capacity of the ironer. If the productive capacity of the ironer is increased in the same proportion as the decrease in moisture content of the *pre-*

TABLE 6—GENERAL VENTILATION RATES FOR APPAREL PRESSING UNITS

PRESSES COMPRISING UNIT	MAXIMUM UNIT BHP	EXHAUST RATE CFM
1. Tapered Apparel.....	1.9	5,200
2. Tapered Apparel..... Yoke	2.22	6,200
3. Tapered Apparel..... Mushroom	2.27	6,200
4. Coverall Body (3).....	2.25	6,200
5. Tapered or rectangular Apparel..... Mushroom (2)	2.64	7,400
6. Rectangular Apparel (2)..... Yoke	4.12	12,000
7. Tapered Apparel (2)..... Yoke	4.12	12,000
8. Tapered Apparel (2)..... Mushroom	4.17	12,000
9. Rectangular Apparel (3).....	5.70	16,000

Note: Average production, 12 lb/hr. One operator per unit.

*conditioned wash*, Equation 3 remains unchanged. The exhaust volume for the prescribed conditions is, therefore, seen to be a function only of the rate of moisture evaporated,  $MD$ .

In the case of flatwork ironers general ventilation might be considered as an alternative to local exhaust ventilation. Consider a typical 2-roll-120 in. and an 8-roll-120 in. ironer rated at 3.8 and 14.2 bhp, respectively. For family flatwork with net identification, the production usually obtained from each of the above ironers would approximate 100 lb per hour for the 2-roll ironer and 400 lb per hour for the 8-roll ironer. The exhaust volumes from Equation 3 for each of these ironers are 6400 and 20,000 cfm respectively.

If canopy exhaust hoods are used with each of these ironers, minimum local exhaust rates of 3000 and 11,000 cfm (Table 1) are required. Assuming these rates exceed those based upon 50 cfm per square foot of net opening, it is quite obvious that general ventilation requirements for ironers will be approximately twice as great as the local exhaust rates listed in Table 1.

In determining the actual design exhaust volumes, it is necessary to consider several factors present in the workroom affecting the removal of heat by the exhausted air. A non-uniform airflow distribution caused by obstructions located

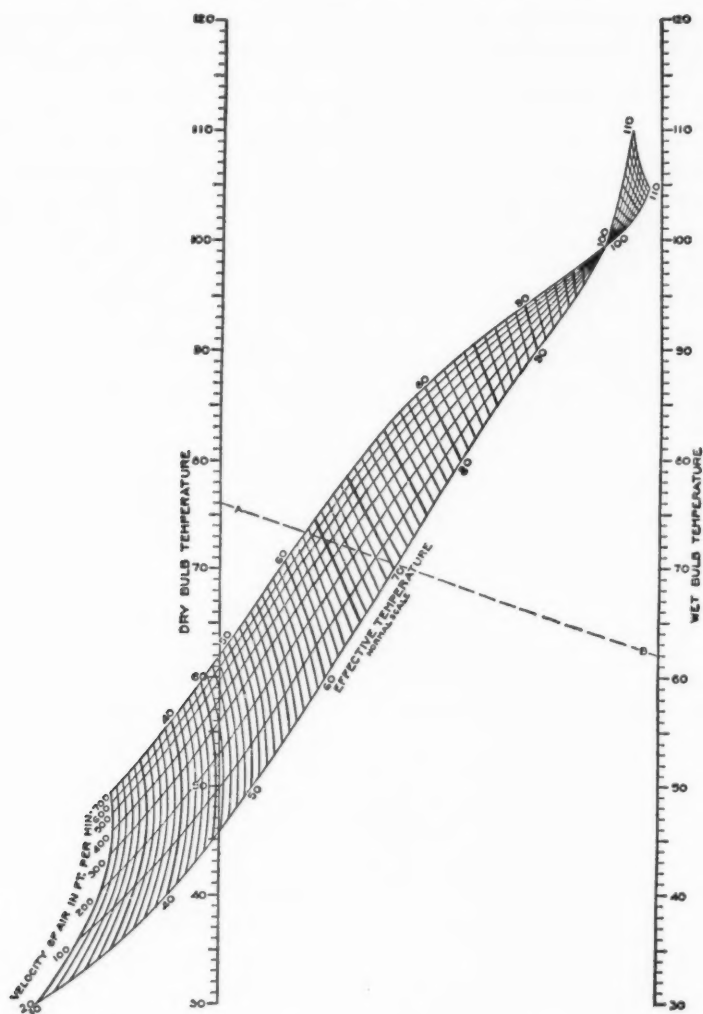


FIG. 3. NORMAL SCALE OF EFFECTIVE TEMPERATURE (APPLICABLE TO PERSONS AT REST AND NORMALLY CLOTHED)

between the fan and the presses, the presence of washers near the press area, additional heat being generated by hand ironing, etc., must be evaluated in terms of a safety factor by which the minimum exhaust volume must be multiplied. It is usually not necessary to exceed a safety factor of 2.

#### EFFECTIVE TEMPERATURE

Effective temperature may conceivably be used as a basis for design instead of the dry-bulb temperature. Theoretically, this would be more desirable. The effective temperature for a given subject is a function of the air movement as well as the dry- and wet-bulb temperatures as shown in Fig. 3. A superficial air velocity would therefore have to be computed from the total exhaust volume and the size of the workroom. The heat load and the amount of moisture released to the workroom are fixed by the type of unit and its productive capacity. It would, therefore, be necessary to proceed by trial and error to choose an exhaust volume that is compatible with Equation 3 as well as with Fig. 3. Since this would be a lengthy procedure which is not warranted by the accuracy of the data describing the conditions in the workroom, it is not recommended.

#### FAN LOCATION

The location of the fan and the points of air make-up to the workroom are the 2 important factors influencing the direction and path of air movement within the workroom. Since it is common practice to have the flatwork finishing and washers

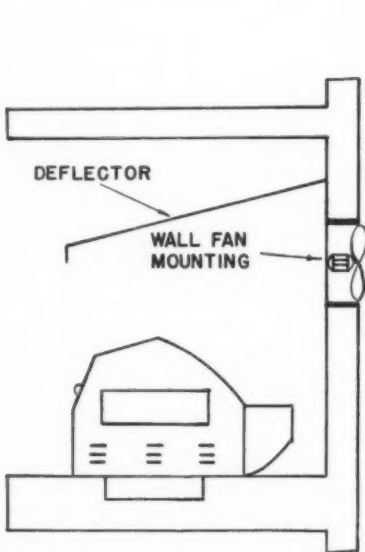


FIG. 4. VENTILATION OF WASHERS;  
WALL FAN MOUNTING

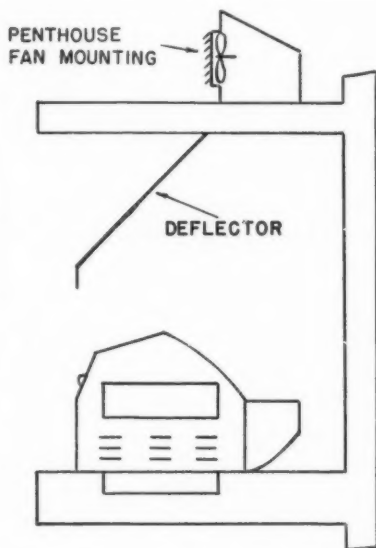


FIG. 5. VENTILATION OF WASHERS;  
PENTHOUSE FAN MOUNTING

located in separate sections, if not separate rooms, air movement must be directed in such a manner as to prevent the hot moist air from mixing with the general room air.

If washers are located in a line along an outside wall, a series of propeller fans mounted in the building wall or windows may be used as shown in Fig. 4. A number of small fans will distribute the air flow better than one large fan. A deflector located above the washers will also improve the air flow pattern. It is also desirable to use outside baffles to prevent the wind from adversely affecting the capacity of the fans. When direct discharge would create a nuisance, stacks discharging above the roof should be provided. If the outside wall is next to an adjacent building or if the washers are located next to an interior partition, air must be exhausted through the roof as shown in Fig. 5. If washers are not located near walls, roof ventilators located above the washer and discharging through the roof are satisfactory.

Press areas may be ventilated in the same manner as the washers. Exhaust fans can be mounted in windows of the walls close to the presses with the bottom of the fans approximately even with the top of the presses. Where this type of exhaust arrangement is not possible, an overhead duct exhaust system with grilled openings located above the press units has been used effectively.

In order to control air movement and prevent *contamination* of makeup air entering the workroom, makeup air inlets should be located between the press

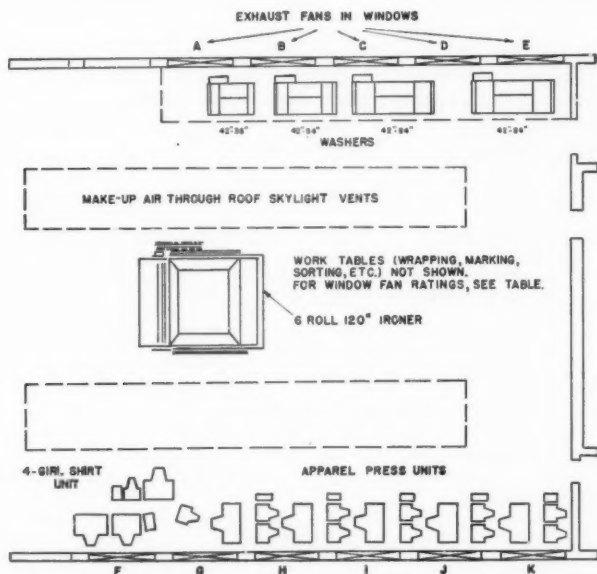


FIG. 6. LAYOUT OF WASHERS, IRONER AND PRESSES IN A LAUNDRY SHOWING LOCATION OF FANS AND MAKE-UP AIR INLETS

and washer areas. The simplest method of doing this, for a one-story building, is through skylight vents located in the roof. This must be taken into account in the choice and in the design of buildings to be used for laundries.

The ventilation rates given in Tables 5 and 6 have been based upon definite outdoor conditions, namely, a dry-bulb temperature of 90 F and a wet-bulb temperature of 75 F. Since the actual outdoor conditions will vary throughout the year, these rates can only be used as a basis for choosing the size and number of fans. The infrequent number of days encountered during the summer months in New York State which might impose more severe conditions does not warrant a more rigid basis. On the other hand, these exhaust rates are greater than actually required during the winter months. It is, therefore, necessary to proceed in the same manner as in air conditioning design; namely, to choose the equipment on the foregoing basis and to adjust the operation of equipment depending upon the outdoor conditions. Thermostatic controls, as used in air conditioning installations, are not warranted in the case of laundries; manual adjustment, by simply shutting off one or several of the fans is satisfactory. In this manner, sufficient flexibility is attained for proper operation throughout the entire year.

#### VENTILATION IN A TYPICAL COMMERCIAL LAUNDRY

To illustrate how these principles are applied, consider the typical commercial laundry shown in Fig. 6. The design requirements for this laundry are tabulated in Table 7.

TABLE 7—VENTILATION REQUIREMENTS FOR A TYPICAL LAUNDRY

LAUNDRY EQUIPMENT INCHES		DESIGN EXHAUST RATES	FAN LOCATIONS & SPECIFICATIONS
Washers	42 x 36	940	Install 3 fans at window locations A, C, D, each capable of exhausting 900 CFM; at location B, 1200 CFM; at E, 1700 CFM.
	42 x 54	1200	
	42 x 84	1700	
	42 x 84	1700	
Total:		5540 CFM	5600 CFM
Ironers	1-120" 6-Roll	8500 CFM (Table 1)	Install canopy hood (Fig. 1). Duct fan for canopy exhaust selected to exhaust 5200 CFM at system resistance.
	Local exhaust		
Presses	1-4 girl shirt unit (#11 from Table 5)	12,000 CFM	Install at window location F, fan capable of exhausting 9,000 CFM; at window of location G, 3,000 CFM. (Total: 12,000 CFM)
	5-Apparel press units (#5 from Table 6)	37,000 CFM (7400 CFM per unit)	
Total:		49,000 CFM	52,000 CFM

## APPENDIX

## FLATWORK IRONERS

To estimate the ventilation rate which would limit the temperature of the exhausted air to 125 F, it is necessary to make a heat balance. Sensible heat is transferred from the ironer rolls to the flatwork and the air passing under the hood. As shown in Table 1, column 4, approximately 65 percent of the delivered bhp is sensible. The remainder of the delivered bhp is dissipated as latent heat in vaporizing the moisture from the entering flatwork.

To simplify calculations, we may assume an average heat capacity of 0.30 Btu per (lb) (F deg) and an average specific volume of 16.4 cu ft per lb for the moist heated air covering the range between 100 F and 125 F. The temperature of the wet flatwork entering the ironer may be considered to be at the wet-bulb temperature of the workroom air. The air entering the workroom is assumed to be at a dry-bulb temperature of 90 F and a wet-bulb temperature of 75 F. Since the temperature rise in the workroom is limited to 10 deg, the workroom wet-bulb temperature will be equal to 78 F. This temperature,  $t_i$ , will be considered the temperature of the flatwork entering the ironer. The temperature of the flatwork leaving the ironer,  $t_o$ , may be assumed to be equal to 150 F. The specific heat of the cotton flatwork,  $s_f$ , may be taken as 0.32 Btu per (lb) (F deg).

A heat balance of the sensible portion of the delivered bhp leads to the following expression:

$$0.65 B [\text{Flatwork bhp}] = 60 (Q/V) s_a \Delta t_1 + D s_f (t_o - t_i) \quad (\text{A-1})$$

After substituting the appropriate values for the terms in Equation A-1 the following is obtained:

$$Q = 794 (\text{bhp}) - 0.84 D \quad (\text{A-2})$$

The second term,  $0.84 D$ , represents the amount by which the ventilation rate must be lowered to account for the heat transferred to and taken away by the flatwork. This term represents less than 3 percent of the ventilation rate and may therefore be neglected. The abbreviated Equation 1 is used to calculate the ventilation rates given in Table 1.

## WASHERS

Consider a washer room completely isolated from the rest of the laundry. The make-up air temperature may be taken as 90 F. If the temperature rise is limited to 10 F, the change in specific enthalpy of the air (assumed to be dry) will be 2.40 Btu / lb. The foregoing assumptions make it possible to make a heat balance resulting in the following expressions:

$$Q = (Uv \Delta t_2) A_s / (\Delta h \times 60) \quad (\text{A-3})$$

Considering the surface of the cylinder to be the effective heat transfer area, we obtain

$$A_s = 0.0218 d [N + (d/2)] \quad (\text{A-4})$$

Combining Equations A-3 and A-4 and substituting the following numerical values,  $U = 3.0$  Btu per (hr) (sq ft) (F deg),  $\Delta t_2 = 60$  F,  $v = 13.4$  cu ft per lb,  $h = 2.4$  Btu per lb, the exhaust volume becomes a function of the nominal size of the washer as in Equation 2.

## PRESSES

Consider an ideal situation where the presses which comprise a unit finishing a complete garment are isolated in a room separate from the other laundry equipment so that material and heat balances may be computed (Fig. A-1).

This system is analogous to a nonadiabatic humidifier with the moisture being added to the entering air from the wet clothes. A heat balance for the system gives

$$q = Gs_1 (t_2 - t_1) + G (H_2 - H_1) [L + c_p (t_2 - T)] \quad (A-5)$$

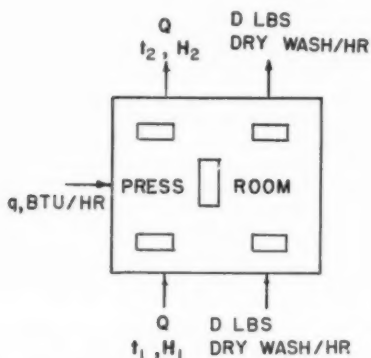


FIG. A-1. HEAT AND MATERIAL BALANCE AROUND A PRESS ROOM

A material balance for the system gives:

$$G (H_2 - H_1) = MD \quad (A-6)$$

Since

$$Q = (Gv/60) \quad (A-7)$$

and

$$q = B \text{ (bhp)} \quad (A-8)$$

Equations A-5 and A-6 may be rewritten as in Equation 3 or these same equations may be further rewritten as

$$H_2 = H_1 + 0.223 (MD/Q) \quad (A-9)$$

For the computations which follow, the entering air is assumed to be at a dry-bulb temperature,  $t_1$  equal to 90 F and a wet-bulb temperature,  $t_w$  equal to 75 F so that the absolute humidity,  $H_1$ , of the entering air as obtained from the humidity chart is equal to 0.0152 lb of water per pound of dry air.

In New York State, Industrial Code Bulletin No. 27 (the Laundry Code) does not permit the dry-bulb temperature in the workroom to exceed the outdoor dry-bulb temperature by more than 10 deg. Therefore, the limiting temperature in the workroom  $t_2$  is assumed, in these calculations, as 100 F. The moisture content of clothing leaving the extractor and brought to the presses may be considered equal to 0.50 lb of water per pound of dry clothing. Actual measurements made at a local laundry substantiate this value.

By using  $D$  as a parameter and holding it constant, and assuming a series of values for  $Q$ , it is possible to solve for  $H_2$  in Equation A-9 for various values of  $Q$ . Knowing  $H_2$  and  $t_2$ ,  $T$  can be obtained from the humidity chart and in turn  $L$  which varies with

T. By substituting the values thus obtained Equation 3 can be solved for  $bhp$ . In this manner, a relationship between  $Q$  and  $bhp$  may be obtained with  $D$  as a parameter.

We may note that the ventilation rates for presses are based on the overall transfer of heat between the press room and the outside. Even though heat is transferred to the work, it is dissipated within the workroom so that the net effect is actually as depicted by Equation 3.

## DISCUSSION

G. M. HAMA†, Detroit, Mich. (WRITTEN): I have read with interest Mr. Marlow's paper on Laundry Ventilation. His determinations for airflow represent considerable work in obtaining data on a large variety of commercial laundry equipment. Mr. Marlow has tabulated the heat loads for a number of laundry machines; by assuming a temperature rise in the exhausted air he has determined the volume of air necessary to remove the heat from each machine in question. This work is an excellent contribution to the field of ventilation in giving a basis for minimum airflow requirements for laundry operations.

In considering the practical use of this data, I believe it would be well to consider the airflows suggested as minimum rates rather than optimum rates. Mr. Marlow's paper presents rates which are calculated to produce cooling in the working room area which will result in a maximum room temperature 10 F. in excess of the outdoor temperature. Inasmuch as the purpose of ventilation for many laundries will be comfort rather than the 10 deg differential, it appears that airflows greater than Mr. Marlow's suggests should be used. The reasons for this are as follows:

1. The effect of radiant heat is not considered in the calculations. High temperature surfaces may produce overheating in the workers even if the air temperatures are low. Aluminum shielding will reduce this radiant heat effect. However if shielding is not provided, additional velocity over the worker may be necessary to offset the effect of the radiant heat.
2. In climates where the outdoor temperatures becomes high, a 5 deg or less increase in workroom temperature over outdoor temperature may be necessary for comfort. Inasmuch as it is not necessary to heat the air such as in winter time makeup air conditions, the moving of larger volumes of air will not be a high cost item.
3. The airflows for the canopy hoods over the flatwork ironers are based on a capture velocity of 50 fpm. This control velocity is for quiet air conditions. In many plants a control velocity of 100 fpm or more may be necessary because of cross-drafts.
4. The paper assumes that all the heat given off by the laundry process in question will be removed by the calculated volume of air as rapidly as it is formed. This may or may not be true. The rate of conductance for thermal convection varies with the shape of the object cooled and the velocity of the cooling stream of air. It may be necessary to increase the volume of air exhausted to achieve the proper rate of heat removal.
5. The typical laundry problem suggests exhaust fans in the windows and makeup air intakes in skylights. Temperatures of air on building roofs are often considerably higher than outdoor air in shady places. If this arrangement is necessary higher airflows would be required. Makeup air from the east or northside of the building would be preferable from the temperature standpoint.
6. Spillage of water, the dumping of wash water on the floor and other wet conditions prevalent in laundries increases the moisture in workroom air. The practice suggested by the author of multiplying the recommended washroom airflow by 2 should be followed.

H. H. REICH, Pittsburgh, Penna. (WRITTEN): The writer has studied with great interest this creditable paper and offers the following comments for such benefit as they might be to members and the author.

† Bureau of Industrial Hygiene, Detroit Department of Health.

With reference to the example worked out in Table 7 and Fig. 6, the total ventilation requirement would be 62,800 cfm. The writer has made a rough floor layout to determine the size of room required for the amount of equipment selected including liberal aisle and working space and has arrived at a size 70 feet  $\times$  43 feet. Assuming a 14 foot ceiling, there would be a gross volume of 42,200 cu ft and, neglecting the volume of the machinery, 90 air changes per hour. This number of air changes would present difficult problems in distribution of make-up air and would result in prohibitive heating costs in winter. As to the latter factor, it would appear that the make-up air should be brought in through tempering coils rather than through skylight vents as illustrated in the article. The writer's design experience in the heating and ventilating of laundries has resulted in considerable less quantities of air for corresponding equipment and correspondingly less number of air changes.

In reviewing the paper to determine wherein the writer's design practice differed from the author's, I have arrived at the following possible sources of excessive ventilation requirements:

*Flatwork Ironers:* This equipment is often purchased with hood and fan. The exhaust rates given by the author appear to be in excess of manufacturers' standards. As an example, one well-known manufacturer exhausts 2400 cfm and uses a  $\frac{3}{8}$  hp fan motor with the hood of a 6 roll, 120 in. ironer.

*Washers:* The equation 2 is derived from an assumed average surface temperature of 160 F. In an average washing formula, if there be such an average, only one 5 min cycle would be at 180 F. There might be 2 cycles totaling 12 minutes at 160 F. The balance of time, approx 48 min, would be at temperatures varying from room temperature to 160 F. The average temperature of 160 F would thus appear to be approx 30 F high.

*Presses:* The ventilation rates are based upon a dry-bulb temperature increase of 10 F. The limiting of the room temperature is, of course, necessary. The writer presents, however, that a higher air temperature rise across the presses can be allowed without appreciably increasing the temperature of the room. It is necessary that exhaust fans draw the air over the presses directly to the outside and, axiomatically that the presses be located as close to the outside wall as practicable. This is essentially the layout of the author's example. According to the data in the article, the exhaust rate for a unit of 2.64 Bhp with 12 lb/hr production would be 7,400 cfm. If a temperature rise of 35 F is allowed, the rate, according to Equation 3, would be approx 2,140 cfm. The 35 F temperature rise was selected only as an example to illustrate that considerable savings in ventilation air could be accomplished by allowing a more practical temperature rise of the air between the presses and the exhaust fans.

J. D. SLEMMONS, Delaware, Ohio: I believe that a lot of us have found in producing some degree of comfort without cooling the velocity of air is vitally important.

When people are working in areas that have large radiant heating surfaces considerable benefit can be had if the air is brought into these rooms directing it upon the people as they are working.

Laundry machinery hoods are large radiating surfaces. It is agreed that large quantities of air must be exhausted from these work areas.

It is however, difficult to control air flow in a room by simply dumping in air and exhausting it out. Ventilation air blown directly upon the body of a worker exposed to excessive heat, both radiation and convection, tends to relieve surface temperatures, increases evaporation with considerable relief to the worker.

Velocities may require some variation depending upon individuals. However, velocities up to 1000 fpm and volumes of 2000 cfm and more are often desirable for each individual.

H. F. ULOVEC, Los Angeles, Calif.: My comments will reflect some of the practical experience we have had in laundry ventilation installations.

Exhaust air velocities and air volume are not the complete answer to comfort work-

ing conditions in laundries. Supplement makeup supply air should be introduced at spot locations through high-velocity outlets directed at the workers around tumblers and mangles. Wherever possible we also try to introduce the supply air just above the floor through plenum chambers below work tables. In connection with press room areas particularly in hotel and institution laundries where the presses generally are closely grouped and ceiling heights are low we have found that the air change ventilation methods are not satisfactory. In such areas we recommend a reduced air change together with mechanical cooling for the supply air.

**AUTHOR'S CLOSURE:** The several questions raised by Mr. Reich are very well taken. I would like to consider each of these points separately.

**Make-Up Air:** The actual size of the workroom, part of which is shown in Fig. 6, is 84 ft long by 60 ft wide and 12 ft high. The volume of the room is approximately 61,000 cu ft. For a total exhaust volume of 63,000 cfm, there would be about 63 air changes per hour. This value, when compared with air change schedules given for various types of rooms and spaces, may appear to be unusually high. However air change can not, in itself and particularly in this case, be used as a design criterion for ventilation. The basis of ventilation for heat removal is essentially *dilution* ventilation, where the (contaminated) heated air is (diluted) cooled by the cooler make-up air. I might add by analogy that in industrial process ventilation for the control of toxic vapors, ventilation control may be achieved through local exhaust or dilution (general) ventilation. Where the latter is used, the air quantities are usually very high. The application of dilution (general) ventilation for control of heat, as in laundries, necessitates the use of large exhaust volumes.

Secondly, the question of heating the make-up air is superfluous. The purpose of providing ventilation is the removal of excessive heat and moisture from the laundry workroom. The ventilation rates given in the paper are based on definite outdoor conditions, namely a dry-bulb temperature of 90 F and a wet-bulb temperature of 75 F. Laundry ventilation designed on this basis will give exhaust rates greater than actually required during the winter months. During the winter it would be necessary to shut off one or several of the fans in order to obtain an adequate heat balance within the workroom. The basis for design given in the article should satisfy the summer period when ventilation is most needed. The infrequent number of days encountered during the summer months in New York State which might impose conditions of greater severity than the design conditions does not warrant a more rigid basis for design.

**Flatwork Ironers:** I do not know the manufacturer of flatwork ironer hoods to which Mr. Reich referred. However, general data supplied by one such manufacturer specifies 16 in and 18 in disc fans for 4, 6, and 8 roll ironers. A  $\frac{3}{4}$  hp motor is specified for driving the fan. The manufacturer rates the fan at 2500 cfm at  $\frac{3}{4}$ " resistance. No attempt is made to base the exhaust volume on the heat load (size) of the ironer. In addition the air volume specified by the manufacturer appears to be *over-rated*. An inspection of the fan rating tables of several fan manufacturers of the type fan that would be used, reveals a maximum rating of 1940 cfm at  $\frac{3}{4}$ " S.P. for an 18 in fan running at 2470 rpm. Unless relatively high pressure vane axial or tube axial fans are used, which is quite unlikely, the specifications set forth by the manufacturer of the flatwork ironer hood appear to be unreliable.

In addition, one of the criteria determining the ventilation rate for laundry flatwork ironers is the temperature rise permitted within the canopy hood. A maximum temperature of 125 F was selected in computing the ventilation rates in Table 1. For these ventilation rates, no prerequisites as to the type of hood construction or insulation was assumed. In cases where double-wall construction and/or other means of hood insulation is used, the ventilation rates given in Table 1 may conceivably be reduced. However, the effect of such insulation is unknown. Therefore, I feel that the ventilation data given in Table 1 may best be used as is.

**Washers:** As pointed out in the paper, the ventilation rates for washers can only be estimated without any great degree of accuracy. The greatest factor of uncertainty

is the effective heat transfer surface area rather than the washer shell temperature. It is quite true that the choice of 160 F or any other temperature between 100 and 180 F can only be described as an *educated guess*. Mr. A. M. Harp who has prepared a Laundry Ventilation Manual for two well-known manufacturers also arrived at an average washer shell temperature of 160 F.

*Presses:* The ventilation rates for presses may be based on any permissible increase in dry-bulb temperature. In New York State, Industrial Code Bulletin No. 27 (the Laundry Code) does not permit the dry-bulb temperature in the workroom to exceed the outdoor dry-bulb temperature by more than 10 deg. The area around the presses where the operator is located is considered a part of the workroom. To permit a higher temperature rise based on 90 F dry-bulb outside make-up air would therefore be inconsistent with the requirements of the New York State Laundry Code.

Both Messrs. Slemmons and Ulovec suggest the use of a separate make-up air system, with the make-up air outlets as close as possible to the workers. Such an arrangement would take advantage of the higher air velocity at such points to produce an added cooling effect. However, a make-up air supply system is usually applicable where an exhaust system is controlling gaseous or particulate contaminants. In geographical areas where heating requirements are considerable—and in buildings that are tightly constructed,—tempered make-up air provided by a separate system is usually the only practical means of balancing the exhaust system. In warmer climates where buildings require little or no heat, natural infiltration through open doorways or windows generally proves to be satisfactory. In the case of laundries, where there is no problem of tempering make-up air, natural infiltration should be adequate.

The ventilation rate of 2000 cfm per individual worker as suggested by Mr. Slemmons does not consider the heat load of the laundry equipment. I therefore cannot see any basis for the use of such a figure.

I must agree with Mr. Ulovec's comment that general ventilation is not the most effective means of controlling excessive heat and moisture in the laundry workroom. The basis for ventilation depends on a differential between the outdoor and indoor temperatures. It is conceivable that in certain geographical locations, or under certain climatic conditions, this differential may prove to be insufficient to lower appreciably the laundry workroom temperature. Air conditioning is the alternative and I might add, more costly.

Mr. Hama states that the purpose of ventilation for laundries should be for comfort. This, I believe, should not be the case. Ventilation is needed not to keep workers comfortable but actually to prevent acute discomfort or physiologic damage due to abnormally hot and/or humid atmospheres. The ventilation rates presented are meant to be minimum rates keeping this purpose in mind.

Since saturated steam is fed to the steam chests and bucks at about 100 psig, the surface temperature of the rolls or bucks would be approximately 340 F. The effect of radiation, when compared with the conduction and convection effects, may therefore be neglected.

Mr. Hama suggests a capture velocity of 100 fpm or more for the flatwork canopy hoods. The 50 fpm mentioned in the paper is a minimum value and should be increased where conditions warrant.



**1547**

## AIR CONDITIONING COIL ODORS

By A. B. HUBBARD\*, BLOOMFIELD, N. J., NICHOLAS DEININGER\*\* AND FREDERICK SULLIVAN\*\*, CAMBRIDGE, MASS.

EXTENSIVE field investigations and thorough laboratory experiments have established a method of predicting the odor performance of cooling coils used in air conditioning. The equipment designer not experienced along these lines may, in choosing coil surfaces, set up a small but troublesome probability of complaints of foul odor easily traceable to the air conditioner discharge grille or duct.

It is necessary at the outset to distinguish carefully between 2 categories of odor trouble. The more common odor complaint is the result of too much tobacco, food, body odor, or combinations of these being produced for the amount of ventilation or infiltration air available to dilute them. The offending odors plainly indicate their origin, such as tobacco, food or body odor. Installers and servicemen are prepared to correct such situations through intelligent appraisal and action. The routine odor difficulty has 2 unique characteristics; the odor is easily identifiable as to origin, and the air-conditioning system is not its focal point. There is no need to elaborate on this ordinary odor problem.

The other type of odor trouble is the subject of this paper and is called by experienced servicemen, *air-conditioning odor*, *coil odor* or *air-conditioning coil odor*. Coil odor is not only very foul and disturbing, but is both foreign and familiar to every layman. It is foreign in that such a blend never occurs in any other connection. It is familiar because coil odor vividly reminds each person of his most revolting odor experience. Different people may describe the same odor situation in terms varying from *stale cigar butt* through *rotten eggs* to *dead rat*. Another distinctive symptom of coil odor is that it clearly comes from the air conditioner. The onset of coil odor takes 1 of 2 forms. It may appear in full force within a week of installation. On the other hand, the early indication may be only a puff of foul odor during the morning start-up. In either case, the premises may be overwhelmed for several weeks during September and October, particularly when a long sustained spell of hot weather suddenly moderates.

A research program, active for almost four years, was started in the fall of 1948. The opening phase of the program was a lengthy field investigation of *store cooler* installations in highly industrial and commercial districts of New York, New Jersey and Philadelphia areas. A team of engineers visited a large number of air-conditioned commercial establishments, mostly selected at random, to obtain first-hand knowledge of odor conditions and to discuss these with users. Due to the difficulty of describing odors, normal channels had generally produced vague and contradictory information. It was felt necessary to build up a solid foundation of facts before a scientific approach could be planned.

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\*\* Senior Research Chemist, Arthur D. Little, Inc.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

## FIELD INVESTIGATION OF ODORS

Early in the field study 2 classes of odor difficulty, as just described, became apparent, and it was clear that effort should be concentrated on further investigation of the coil odor phenomenon. The survey revealed 13 installations having coil odor and to which experimental access could be arranged.

The selected coil odor cases had in common varying degrees of tobacco, food and body odor as suspected primary sources from which odor accumulation could be drawn. All these units had bare copper coils. Although bare copper had been used successfully for over 15 years, it was decided to test the persistent rumor that copper accumulates foul odor. One store cooler was selected for experiment. This unit was in a night club with year-round cooling load, and there were competitive units, with different coil surfaces, on the same premises. Highly thinned red alkyd base paint was flow-coated on the copper coil surface in place. When improvement was noted, the remaining installations selected for experiment were similarly treated one at a time as justified by preceding experience. Results proved fair to excellent.

The investigating team of engineers inspected each of the experimental jobs at monthly intervals for a period of 1 year. Visits to other installations were also kept up to provide general experience and a basis for comparison.

Various approaches were made to determine the mechanism of coil odor. For example, numerous samples of condensate were collected and analyzed. Parts per million concentrations of many contaminants were noted. These included sulfates, nitrates, nitrites, chlorides and phenol. Some of these correlated fairly well with coil odor conditions. However, glaring exceptions were evident, and no individual condensate analysis could be used to predict susceptibility to coil odor.

After a year of experience, the field investigating team had accumulated a store of empirical know-how, but found itself in an unsatisfactory situation. Odor trouble could not be accurately predicted, and the ability to diagnose air-conditioning odor and prescribe corrective action could not be readily transferred to others. No positive conclusions of importance could be stated. For instance, it could only be said that a very small percentage of bare copper coils would develop odor; the majority were acceptable, and many were excellent. It could not be said that tinned copper coils were immune to odor; investigation had turned up some aggravated cases of odor on such coils. Coil odor is unknown in some areas and fairly common in others. Some establishments, such as a particular delicatessen-meat market, had a high level of odors of all varieties, but no signs of coil accumulation. On the other hand, there was the unlikely case of coil odor in an appliance showroom with no discernible background odor except from smoking during infrequent sales meetings.

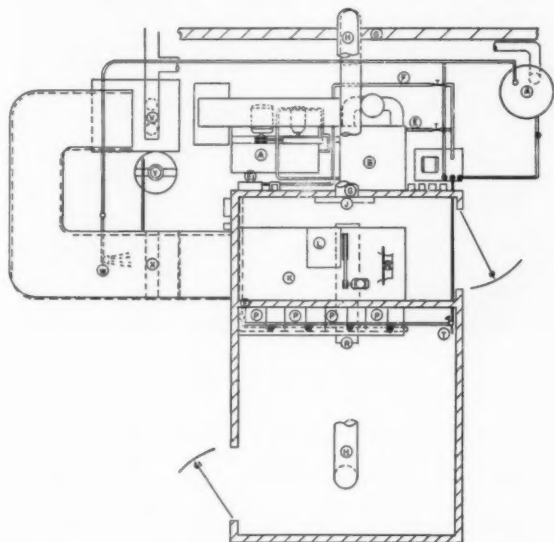
Thus it was apparent that the research program should go into a laboratory phase where scientific methods and controls could be used to clear up the contradictions and answer the questions raised by the field study. It was further apparent that the objective of a laboratory program should be to reveal the mechanism and fundamentals of coil odor. Arrangements were made to take up the scientific phase in an Odor and Flavor Laboratory in Cambridge, Mass., and the program called for close cooperation between the field study team and the laboratory engineers and scientists with related odor skill and experience.

The first move was to design and construct a test room where 4 conveniently small coils, having different surfaces, could be operated under controlled odor

input conditions. It was then necessary to adapt a suitable method of odor measurement, to develop test procedures and to analyze data to rate the odor properties of various cooling coil surfaces.

### ODOR TEST ROOM

The odor test room, constructed of asbestos-cement board, had overall dimensions of  $8 \times 12$  ft with a  $7\frac{1}{2}$ -ft ceiling. The room was divided into two chambers;



(A) F-12 Condensing Unit. (B) Cooler. (C) Circulation Pump. (E) Water Inlet. (E1) Water Outlet. (F) Chiller By-pass. (G) Outside Air Intake Duct. (H) Air Exhaust Duct. (J) Glass Fiber Filter. (K) Mixing Chamber. (L) Blower. (N) Temperature and Humidity Recorder. (P) Experimental Coils. (R) Air Recirculation Duct. (S) Water Manifold. (T) Water Pressure Gauge. (U) Gas-fired Water Heater. (V) Gas-fired Furnace. (W) Water Spray. (X) Capillary Cell Humidifier. (Y) Condensate Pump.

FIG. 1. PLAN VIEW OF TEST ROOM

one chamber ( $8 \times 4 \times 7\frac{1}{2}$  ft) served as the odorant introduction room; the other chamber ( $8 \times 8 \times 7\frac{1}{2}$  ft) contained the test coils and was used as the panel examination room. Fig. 1 is a top-view drawing of the test room with apparatus and equipment labeled. The smaller of the 2 chambers of the test room is used for the direct introduction and mixing of the specific odorants and outside air prior to circulating the mixture over the coil surfaces. The odorants selected were tobacco smoke, cooking food odors, synthetic perspiration sprays and combinations of these. These odorants had been suggested by the Technical Advisory Committee on Odors of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS as being the most likely offenders in air-conditioning systems.



FIG. 2. INTRODUCTION OF FOOD ODORS IN THE SMALLER CHAMBER OF THE TEST ROOM

Fig. 2 shows the method of introducing food odorant. Fig. 3 shows an arrangement of 4 experimental coils in the panel examination room.

In this test room it was possible to reproduce coil odors in a standardized way which almost exactly duplicated actual conditions observed in the field. The characteristic behavior of test coils made of different metals and having different surface treatments could be readily evaluated. The test room made it feasible to study the accumulation and modification of specific odorants on the experimental coil surfaces during the cooling period, the subsequent release of odor components with an increase in coil temperature and the extent of odor retention after a return to initial cooling temperature.

#### ODOR MEASUREMENT

Odor evaluations were made by a trained panel using an adaptation of the Flavor Profile method of Cairncross and Sjostrom<sup>1</sup>. Each panel member smells the air leaving the coils and writes down odor notes perceived, using descriptive terms such as *sour*, *burnt*, or *rubbery*; and decides on the intensity of each note using a number scale ranging from *threshold* (just perceptible) to *very strong*. Findings are discussed fully by the panel in open session after each test until all agree on common terminology. The resulting consensus is an *odor profile* for each coil. After considerable experimentation, the following cycle was established:

<sup>1</sup> Flavor Profiles—A New Approach to Flavor Problems, by S. E. Cairncross and L. B. Sjostrom (*Food Technology* IV (8), 1950, p. 308).

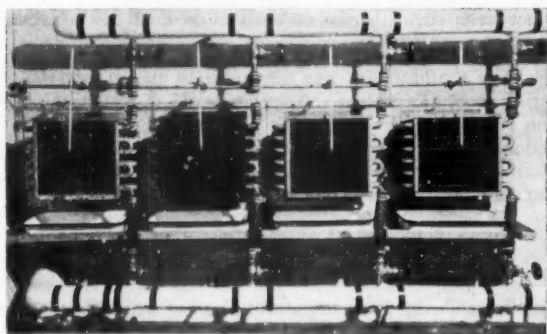


FIG. 3. FOUR EXPERIMENTAL COILS AS VIEWED BY OBSERVER IN THE PANEL EXAMINATION ROOM

1. Normal operation: chilled water is circulated through coils, and outdoor air is circulated through rear chamber, coils and then back to outdoors.
2. Odorant is introduced in rear chamber while chilled water is circulated through coil and test room air is recirculated. Water temperature is adjusted to give 60 F leaving air temperature, and odorant introduction is continued 15 min.
3. Rooms are flushed with outdoor air for 15 min while chilled water circulation is continued.
4. Odor measurements are made by panel immediately after flushing period and while outdoor air flushing is continued. Fig. 4 shows a panel examining 4 experimental coils.
5. Coils are purged of odor accumulation by switching water circulation to stored hot water with temperature adjusted to give 100 F leaving air.
6. Odor measurements are made at the beginning of the purging period and repeated after two 10-min periods.
7. Odor retention is determined by making a final odor measurement after chilled water circulation is re-established and leaving air temperature is again 60 F.

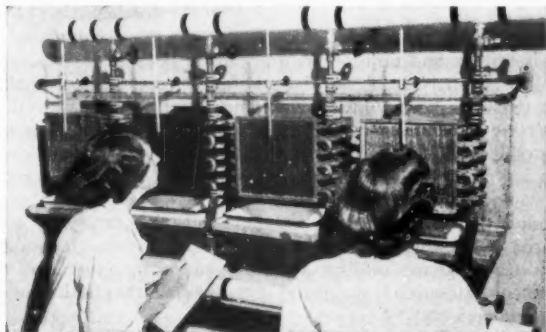


FIG. 4. PANEL MEMBERS TESTING EXPERIMENTAL COILS IN EXAMINATION ROOM

8. Finally hot water circulation is continued to drive off any remaining odors and to prepare coils for another test.

During the tests the humidity was controlled to insure that the coils dripped normally during cooling operation. A special series of tests established that dry coil operation resulted in the least odor pickup, normal dripping operation increased the odor load, and *borderline dehumidification* caused a noticeable worsening of odor performance. This last operating condition is characterized by a heavy coating of moisture on the coil surface with drops clearly seen but with none dripping off.

#### EXPLANATION OF COIL ODOR

Air conditioning coil odor is an unnatural blend of odorous substances accumulated on the surface of the coil. It is caused by selective adsorption of various



FIG. 5. PANEL EXAMINING TEST STRIPS

odor components from the normal environmental odors. Some components of a complex odor blend are adsorbed more readily than others, the speed of adsorption and the preference patterns being functions of the kind of surface. An impressive amount of odorous material may collect on a coil surface even though the environmental odor is below or near threshold level. Adsorption is more rapid on a cold surface and is probably influenced by the condensation of water vapor.

When released by the heating or off-cooling cycle, this accumulated odor residue (coil odor) is foreign and objectionable primarily because it is out of context or separated from its normal blend of background odors. The emphasized components of the odor accumulation are generally the more objectionable odors because they are associated with putrefaction, rancidity or uncleanness.

Common odor blends, for example, tobacco, food and body odors, are not as different as one might suppose. Such notes as burnt-fragrance, sour, sulfide, and animal are common to all three of these odorants. The similarity of tobacco and vegetable odors can easily be demonstrated in the laboratory by comparing the odor produced by passing cigarette smoke and steam through copper wool, with the odor of steam passing through a mixture of cut-up vegetables. The identifying component of tobacco is scalped by the copper so that both appear as familiar boiling vegetable odors.

The fact that coil surfaces do selectively adsorb odors from the air passing over them introduces complications in the analysis of test results. The odorant blends used include at least 30 components or notes, and each kind of surface may pick

up any number of these in amounts above threshold. Each picked up component varies in strength depending on the preference pattern of each surface. The acceptability of each note picked up is another variable. Finally, each surface, when warmed, exhibits a different ability to retain each adsorbed odor component. The Appendix outlines the method used to reduce test results to predictions of comparative quality of surfaces to air-conditioning cooling coils.

#### RAPID SCREENING OF SURFACES

The cost of making and testing actual coils turned out to be too high to permit speculative search for better surfaces. To overcome this, a simple rapid test was devised. Briefly, the screening test consisted of the following steps:

1. Initial examination of 4-in. sq test strips by a trained panel.
2. Contamination by tobacco smoke in saturated air for 1 hr followed by 5 min airing.
3. Odor profiles of contaminated strips by panel at room temperature.
4. Odor profiles of contaminated strips during gentle heating by infrared lamp, special attention being paid to the rate of purging.
5. Final odor profile after cooling contaminated strips to room temperature.

Whenever possible the test strips were baked at 250 F for 4 hrs before being contaminated.

The odor profile procedure has already been outlined. Fig. 5 is a photograph of a typical panel at work. The panel operates in an odor-free room without distraction. For maximum efficiency, each panel member must be completely at ease and confident. The panel leader has the delicate and demanding task of guiding the panel without dominating its judgment or upsetting its morale and confidence.

Each material was rated on the basis of its odor pickup resistance. This was a term used to incorporate both the type and level of odor pickup. The rating scale was divided into four distinct categories, *good*, *fair*, *poor* and *very poor*. For example, if the odor level was low and the type of odor not objectionable, the surface received a *good* rating. Conversely, a surface exhibiting objectionable components at a high level merited a *very poor* rating. The rate of release or the speed of purging the adsorbed odor was also considered to be of major importance. Those materials which rapidly returned to their original odor during the purging period and, therefore, had little or no residual odor were given a *fast* rating; those which lost only a small part of the acquired odor during purging and retained a residual odor of high intensity were given *slow* ratings. Purging between the two extremes was classed as *moderate*.

Deiningering and Sullivan<sup>2</sup> have described the method in its general application to rating surfaces on odor performance. The description and discussion here covers the special case of rating surfaces with regard to their odor performance in air-conditioning cooling coils.

Fig. 6 shows the apparatus set up and ready to use. The odorizing chambers are standard 10-in. laboratory vacuum desiccators. Each vessel is cleaned and charged with 100 ml of water to provide a saturated atmosphere. Four or five test strips, preferably 4 in. square and about 0.025 in. thick, are suspended on a

<sup>2</sup> Study and Evaluation of the Odor Properties of Surfaces, by Nicholas Deiningering and Frederick Sullivan (Annals of the New York Academy of Sciences, Vol. 58, Art. 2, 1954, p. 215).

glass rod. Each chamber is evacuated to 15 mm of mercury. Five cigarettes of assorted brands are placed in a manifold and smoked at a slow and steady rate down 2 in. by controlling the pressure drop across the cigarettes at  $\frac{1}{2}$  in. of water indicated on a manometer. To insure a strictly vapor phase odorant, the smoke is led to the odorizing chamber through an impingement tube containing a plug of clean non-absorbent cotton. The remaining vacuum is broken by introduction of clean air. The 1-hr contamination period starts at this point.

### RESULTS

Table 1 is a plot of resistance to odor pickup versus purging rate. Surfaces of equal pickup appear on diagonals and of equal purging rate in columns so that the horizontal rows show equal quality coil surfaces. The ideal surface would either

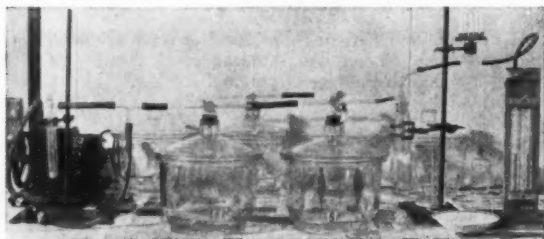


FIG. 6. APPARATUS FOR CONTAMINATING TEST STRIPS

pick up nothing or would avoid any release of accumulated odor. While such an ideal is unlikely, it is reasonable that the best obtainable actual surface would sparingly pick up the more acceptable odors and would release these rapidly during off-cycles to avoid undue build-up. This is the *good* and *fast* area appearing as Class 1 in the top row.

Coil quality in Class 2 is exemplified by *fair* pickup in combination with *fast* purging. Another coil of *moderate* purging rate would have to improve its other attribute, pickup, to *good* in order to stay in Class 2. This is reasonable and has been checked repeatedly. Classes 3 and 4 each have three combinations of pickup and purging; Class 5 has two, and Class 6 has one. (Words fail to convey an idea of the poor quality of Class 6 surfaces which avidly pick up the worst stench and release them interminably.)

Table 1 bears out the old industry rumor concerning bare copper which appears in Class 5 (*poor* pickup and *slow* purging). The same rumor has it that tin coating is the remedy, and this is so to the extent that it promotes copper to Class 4. This finding has been checked with pure sheet tin and also solder coatings on copper.

Anodizing copper (electrolytically induced oxide) degrades it to bottom place along with silver, tungsten, nickel and zinc.

Organic coatings change the odor performance ratings, sometimes for better, other times for worse, but no decisive trend is apparent.

No really practical treatment could be found to raise copper to a significantly higher class. Copper can be upgraded to Class 2 by an expensive wax coating

(commonly used to reduce adherence of ice cubes to aluminum trays). The same result can be obtained at considerable cost by a polyvinyl formal coating which requires an oxide substrate.

Noble or exotic metals do not necessarily give odor performance in proportion to their price. Following are some examples: tantalum, platinum and stainless steel in Class 2; molybdenum and titanium in Class 4; magnesium and cadmium-silver in Class 5; and silver, tungsten, nickel and zinc in Class 6.

PACING RATE			COIL QUALITY
FAST	Moderate	Slow	
CHROMATIZED ALUMINUM (14.5-5-10) PHOSPHATIZED ALUMINUM PAINTED STEEL (POLYVINYL BASE PAINT)			1
CHROMATIZED ALUMINUM (14.5-5-3) ALUMINUM - POTASSIUM DICHROMATE CHROMATED PHENOLIC PHOSPHATE-CHROMATE CONVERSION COATING DOWNE ANODIZE CYCLIZED RUBBER COPPER - POLYVINYL FORMAL/CO SYNTHETIC MALL 2 TANTALUM -	BUFFED ALUMINUM ELECTROPOLISHED ALUMINUM STAINLESS STEEL (18-8) PLATINUM		2
ALUMINUM - CHROMATED PAINT SHELLAC & CHROMIC OXIDE POLYETHYLENE COPPER - CELLULOSE ACETATE/CO POLYETHYLENE/CO PRIMER "A" - POLYVINYL BUTYRAL STEEL - PAINT "A" ASPHALTIC PAINT	ALUMINUM - FIN STOCK (555) CHROMIC ACID ANODIZE PRIMER "A" POLYESTER CHROMATIZED (14.5-5-15) PAINT "B" COPPER - EPOXY RESIN POLYESTER TANTALUM (VAPOR DEP.) POLYVINYL ALCOHOL/CO NEOMETHYLENE GLARINE ADIPATE STEEL/CO STEEL - EPOXY RESIN		3
ALUMINUM - POLYTETRAFLUOROETHYLENE (ELECTRODEPOSIT)	ALUMINUM - PHOSPHATE PRIMER "A" - POLYVINYL BUTYRAL RUBBER HYDROCARBON - PHENOLIC POLYVINYL CHLORIDE COPPER - SILVER PLATE + SILVER JODINE SHELLAC (ELECTRODEPOSIT) PRIMER "A" STEEL - PAINT "A"	ALUMINUM - OXIDE FILM $(\text{Co}(\text{OH})_2 + \text{CaSO}_4)$ POLYTETRAFLUOROETHYLENE EPOXY RESIN OXIDE ANODIZE (ETCH) ICE CUBE TRAY STOCK COPPER - CHROMIC PLATE FIN (ELECTROPLATE + "PLANT") STEEL - FIN STOCK (14M CARBON) POLYESTER TITANIUM -	4
ALUMINUM - POLYVINYL BUTYRAL COPPER - POLYVINYL CHLORIDE STEEL - PHOSPHATE AEROSOL COAT BOARD -	ALUMINUM - AEROSOL OXIDE CONVERSION COATING BOAT ACID ANODIZE VITRIFIED GLASS COPPER - FIN STOCK CHROMIC PLATE SILVER PLATE SHELLAC-CHROMIC OXIDE NEOMETHYLENE GLARINE ADIPATE CADMIUM - SILVER ALLOY		5
	ALUMINUM - TANTALUM (VAPOR DEP.) MOLYBDENUM (VAPOR DEP.) CADMIUM SULFIDE (VAPOR DEP.) SILICON DIOXIDE (VAPOR DEP.) GRAPHITE AEROSOL COATING (HARD) COPPER - ANODIZED POLYTETRAFLUOROETHYLENE POLYTETRAFLUOROETHYLENE SILVER TUNGSTEN - NICKEL - ZINC -		6

TABLE 1—ODOR PERFORMANCE OF SURFACES

Aluminum can be upgraded one class by buffing or electropolishing. This is of academic interest only, since such surfaces would not retain their polish in service. The best improvement of aluminum was obtained by mild chromic or phosphoric acid treatment as explained in detail later.

The foregoing results apply to strips tested with cigarette smoke. It should be of interest to compare these with the results of coils tested with tobacco, food and synthetic perspiration odors. Table 2 shows the odor performance of test coils following the same scheme used in Table 1 for the strips. The situation is more complicated because in many cases the coils have fins of one material and tubes of another. Where direct comparisons can be made the results match except in the case of copper. In Table 2 this shows up in Class 4 instead of Class 5, probably due to some fault in the method of reducing the voluminous coil test data to this form.

Direct comparisons of all-aluminum, aluminum fins on copper tubes, and all-copper coils indicates that they rank in that order from best to worst. That is, the effect of the copper tubes *shows through* to partly offset the better performance of aluminum.

The chromitized and phosphatized aluminum coils fulfill the promise indicated by the strip tests. The copper-oxidized coil produced unbelievably foul stench, again predicted by the screening test.

The inclusion of the two vitreous enamel-coated coils ought to be explained. In the early stages of the project some thought was given to constructing glass coils which could perhaps serve as standards of excellence. When no practical way could be found to design a glass coil of reasonable proportions, it was suggested that vitreous enamel might form an essentially glass surface. It can be seen that such enamel either lacks the odor properties of glass or that glass does not guarantee good odor performance.

*Chromitized and Phosphatized Aluminum;* These acid dips are ordinarily used as mild etches to improve the adherence of paint to aluminum. It was found that such dip treatments improved both pickup and purging performance of aluminum. The three main variables of the process are bath temperature, dip time, and acid concentration. These are noted parenthetically in Table 1 in the order mentioned in degrees Fahrenheit, minutes, and percent. The treatment seems critical and, if overdone, the odor improvement may be lost. Several combinations were tried with the various results shown, but the possibilities were not completely explored.

Special tests were made on a group of coils over an extended operating period to determine the life of treated compared to untreated aluminum coils. No significant loss of odor performance occurred in the equivalent of one cooling season.

Many of the coatings and treatments tested in strip form cannot be applied to finished coils, but must be applied to fin stock ahead of fabrication. Chromitizing and phosphatizing, however, are readily accomplished after the coil is completely assembled.

*Incidence of Coil Odor;* Correlation of field observations and laboratory results permits only the qualitative conclusion that the chances of occurrence of coil odor increase as the class number, according to Tables 1 and 2, increases. No case of coil odor in Classes 1 to 3 has been encountered during five seasons of observation. Previous experience with coils in Classes 4 and 5 indicated less than 2 percent incidence in some areas and negligible occurrence in others. There is no field experience with Class 6 coils.

PURGING RATE			COIL QUALITY
FAST	MODERATE	SLOW	
CHROMA- TIZED/A/A PHOSPHA- TIZED/A/A			1
FAIR	EPOXY RESIN /A/C	GOOD	2
ALKYD BASE PAINT/C/C	A/A PHOSPHA- TIZED /A/C CHROMA- TIZED/A/C VITREOUS ENAMEL/A/C	GOOD	3
VERY POOR	POLYVINY- LIDENE CHLO- RIDE/A/C A/C SOLDER/C/C C/C	FAIR	4
	POLYTETRA- FLUORO- ETHYLENE/A/C SHELLAC/A/C VITREOUS ENAMEL/C/C	CUO PLUS POLYVINYL FORMAL/C/C PRIMER /C/C	5
NOTE: COIL DESCRIPTIONS ARE CODED: COATING/FIN/TUBE A = ALUMINUM C = COPPER		CUO/C/C PRIMER "B" /C/C	6

TABLE 2—TEST COIL ODOR CHARACTERISTICS

It was impossible to predict which installation would develop persistent coil odor and which would be free of it. Laboratory tests gave the broad separation previously outlined but could not duplicate the critical region between susceptibility and resistance to coil odor. The following factors appear to determine between them the probability of coil odor:

1. Concentration and type of environmental odor. Depends on relation between rate of production of odor and ventilation to dilute odors.
2. Kind of coil surface.
3. Ratio of cooling to off-cycle operation *while fan is running*. This depends on relation between load and capacity, and method of operation by user of equipment. The coil with more off-cycle time is less likely to develop odor.
4. Dehumidifying load. Coil odor is more probable if apparatus dew-point remains in borderline zone for long periods. The coil that is wet but not dripping picks up the most odor.

There are undoubtedly additional contributing factors, but to include these would stretch the speculation beyond reason. The geographical pattern fits the theory

in the sense that load is steadier and capacity margins are smaller in seaboard climates. In consequence, one would expect less opportunity for these coils to unload accumulated odors.

### CONCLUSIONS

Objectionable odor level in air-conditioned spaces due to high rate of odor production coupled with poor ventilation is distinguished from the case where the cooling coil surface is the focal point. Coil odors are fairly rare but produce devastating customer complaints, as the odors are most obnoxious and unrelated to normal odors produced on premises.

Laboratory experiments with test coils subjected to controlled contamination and with test strips contaminated by tobacco smoke in closed containers show that a surface selectively adsorbs components of a familiar but complex blend of odors. The adsorbed portion of the original blend is released at possibly higher concentration during off-cycles.

Each kind of surface exhibits a different ability to pick up odors and has a different preference for components. Surfaces also have different abilities to retain adsorbed odors. The best coil surface is one which sparingly picks up the more acceptable odor components and which releases them most readily to avoid build-up of foul odors.

Aluminum, tinned copper, and bare copper coils ranked in that order. Chromic and phosphoric acid dip treatments of aluminum gave the best odor performance of the large number of metals and coatings tested.

An inexpensive, rapid method of screening surfaces was developed and results were shown to correlate with those of coils tested in the laboratory. In turn, these findings were in good agreement with field observations.

### ACKNOWLEDGMENTS

The authors are grateful for the guidance and counsel of E. C. Crocker, S. E. Cairncross, L. B. Sjostrom and E. Kreidl of the Arthur D. Little, Inc., staff. The project could not have been carried out without the foresight and sponsorship of S. J. Levine and C. L. Fay, General Electric Air Conditioning Division.

## APPENDIX

### DERIVATION OF COIL QUALITY RATINGS

A simplified example will be used to illustrate in outline how panel observations are analyzed to predict odor performance of surfaces, and may also emphasize the mechanism of selective adsorption.

Suppose surfaces are exposed to an odor blend consisting of only 3 components. Further suppose that the panel agrees on the quality of the 3 odors, respectively, as *acceptable* (to nearly everyone), *doubtful* (depending on personal taste), and *unacceptable* (to nearly everyone). The panel is instructed not only to identify odors picked up by surfaces, but also to decide whether they are *weak* or *strong*. The 2 factors, quality and strength, combine to determine pickup ratings. The other half of the coil odor phenomenon depends on speed of desorption. Cooling coils tend to purge themselves during off-cycles. It can be seen that the faster this occurs the less the chances that accumulations will be carried over to *build-up*. Accordingly, the observers note the

speed of purging when the samples are gently heated in a standardized way. Observations are recorded as *fast*, *moderate* and *slow* to provide the third factor needed.

The 3 factors can be combined 18 ways to form 3 coil quality ratings. Figs. A-1 to A-4, inclusive, show this graphically in 4 steps.

The 18 surfaces in Fig. A-1 start clean and have no odor. However, they are each different, having been chosen to cover exactly the range of possibilities in the simplified example.



FIG. A-1. EIGHTEEN DIFFERENT SURFACES COULD THEORETICALLY BE CHOSEN TO ILLUSTRATE THE FULL RANGE OF ODOR PERFORMANCE

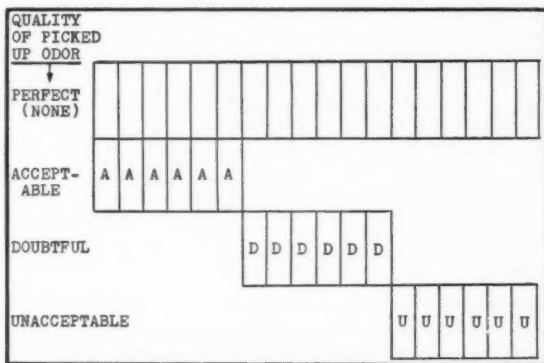


FIG. A-2. AFTER EXPOSURE TO A THREE-COMPONENT BLEND, THE SURFACES DIVIDE THEMSELVES INTO THREE GROUPS BY SELECTIVELY ADSORBING A SINGLE ODOR COMPONENT. THE GROUPS ARE RESPECTIVELY DEMOTED ONE, TWO, AND THREE PLACES FROM PERFECT FOR PICKING UP ODORS RATED AS ACCEPTABLE, DOUBTFUL, AND UNACCEPTABLE

The effect of quality of picked up odor is shown in Fig. A-2. It is necessary to suppose that each surface selectively adsorbs a single component. It is unlikely that any surface can retain a perfect rating by failing to adsorb any odor. Samples are demoted from the perfect rank according to quality of odor. Those having *acceptable* odors are demoted 1 rank on the basis that even these least obnoxious odors are not really desirable. *Doubtful* and *unacceptable* odors are respectively placed 2 and 3 places down from perfect.

In Fig. A-3, samples that adsorb strongly are demoted an additional rank, while those with weak pickup retain the places shown in Fig. A-2. This completes the pickup rating.

Samples with *fast* purging rates are allowed to keep their previous ranks as shown in Fig. A-4. Those with *moderate* rates go down 1 more rank, and those that are *slow* are demoted 2 places below the rank they had in the preceding figure. Predicted performance, or coil quality, now appears on 6 levels lower than *perfect*.

### DETERMINATION OF ACCEPTABILITY CLASSIFICATIONS

The foregoing example indicates the strong influence of odor quality, as compared to odor intensity, in predicting the suitability of a surface for cooling coil application. Field observations clearly support the decision to include odor quality as a prime factor in odor performance ratings. People actually prefer no odors in connection with air

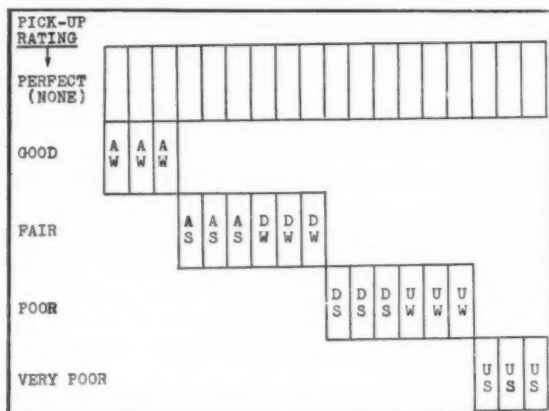


FIG. A-3. PICKUP RATING IS COMPLETED BY CONSIDERING ODOR STRENGTH. SURFACES WITH WEAK (W) ODOR REMAIN IN PLACE WHILE THOSE WITH STRONG (S) ODOR ARE DEMOTED ONE ADDITIONAL RANK

conditioning. They will, however, tolerate rather high levels of some odors while objecting strenuously to others that are barely discernible. There is a middle class of odors which some individuals can not tolerate, but to which others are relatively indifferent. It may be of interest to outline the method of classifying odor notes encountered in this investigation.

Each panel member was asked to list the odor notes in 3 groups on the basis of individual personal reactions. These listed as *acceptable* were felt by the individual to be tolerable (at moderate intensities) in any air-conditioned space; those called *unacceptable* would be intolerable anywhere, and the *doubtful* ones might be objectionable in 1 context but not in another. Lists were compared and a composite made up. Only unanimous choices remained in the *acceptable* column; all nominations for *unacceptable* were included in the composite, and the remainder were classed as *doubtful*.

A more objective method of measuring acceptability would be desirable. However, an opinion poll of a cross-section of the population is not practical, because each person polled would have to be exposed to the actual odors. A completely verbal method of determining odor opinion is not available.

Table A-1 is a record of the grouping used in this study and also indicates the nature and variety of the odor components observed on test strips exposed to cigarette smoke.

TABLE A-1—CLASSIFICATION OF ODOR PROFILE NOTES<sup>a</sup>

ACCEPTABLE	DOUBTFUL	UNACCEPTABLE
Cement Dusty Metallic Papery Straw-like Sweet Fragrance Woody	Burnt Coal Dust Fruity Leathery Medicinal Painty Seaweed Soapy Solventy Sour Waxy	Animal Burnt Fragrance Decayed Vegetable Earthy Food-like Footy Pungent Rubbery Sour (B.O.) Sulfide Tobacco

<sup>a</sup> The order of the notes in each column is alphabetical, rather than indicative of relative acceptability within the groups.

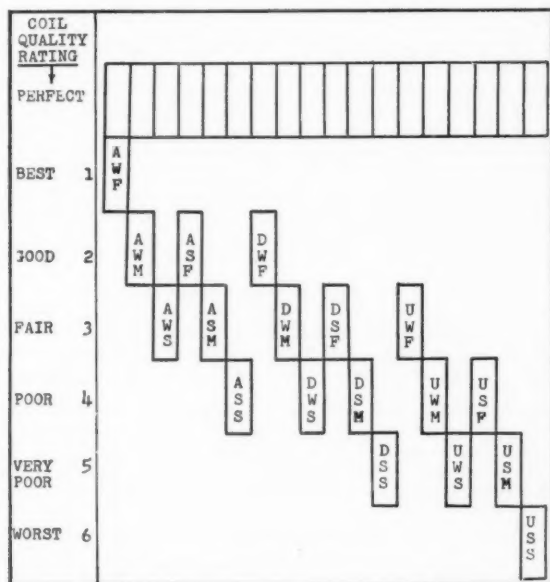


FIG. A-4. ODOR PERFORMANCE IN SERVICE IS PREDICTED BY COMBINING PICKUP RATINGS AND PURGING RATINGS. THE LATTER IS JUDGED BY SPEED OF DESORPTION DURING HEATING AND RECORDED AT FAST (F), MODERATE (M) OR SLOW (S). SURFACES ARE DEMOTED ONE ADDITIONAL PLACE FOR MODERATE, AND TWO FOR SLOW PURGING

## REDUCING ACTUAL RESULTS TO COIL QUALITY RATINGS

The example shown graphically in Figs. A-1 to A-4 was purposely made simple by assuming that only 1 odor component be selectively adsorbed on each test strip. In the actual case several odor notes are adsorbed, and these may have different strengths and acceptabilities. Also, the speed of purging may be different for each component. However, the principle of successive demotions still applies and can be implemented arithmetically.

Figs. A-2, A-3 and A-4, retaining the simplified example for the moment, can be expressed by the following equation:

$$\text{Coil Quality} = q + (s - 1) + (p - 1)$$

where

$q$  = odor quality with values of 1, 2 and 3, respectively, for *acceptable*, *doubtful* and *unacceptable*.

$s$  = odor strength with values of 1 and 2, respectively, for *weak* and *strong*.

$p$  = purging rate with values of 1, 2 and 3, respectively, for *fast*, *moderate* and *slow*.

This equation will yield coil quality ratings from 1 to 6 (best to worst order) with maximum contributions of 3, 1 and 2, respectively, from odor quality, odor strength and purging rate.

In the case of several odor notes appearing on a test strip, coil quality for each note can be determined separately and averaged. Overall coil quality can be taken as the nearest whole number.

Decisions must be made before strength and purging rate can be reduced to numbers. The intensity scale used by the panel might be divided into weak and strong so as to include approximately half the cases encountered within each. Purging rate is indicated by the difference in intensity noted in the first observation after exposure and the final reading after purging. Similarly, this range might be divided into 3.

The main point is that a systematic treatment of data is essential. An extensive investigation produces voluminous information that is difficult to reduce by intuitive methods.

## DISCUSSION

L. H. BECK†, Ann Arbor, Mich. (WRITTEN): The study is excellent. I can foresee many future studies which will use theirs as a prototype. The authors are to be congratulated on their work. It is well-conceived, well-executed, and well-presented.

G. W. MEEK, New York, N. Y.: This paper is without doubt in the writer's opinion one of the finest pieces of practical research that has yet been done by any members of the air conditioning fraternity on the subject of odors. Having at least passing knowledge of the almost endless difficulties involved in even the simplest odor research problem, I can but applaud the scholarly and practical approach taken by Mr. Hubbard and his associates.

A few years ago when the TAC on Odors was organizing its suggested research program one school of opinion wanted all of the emphasis placed on what it called *quick and practical* research. If any proponents of that school of thought still exist, this paper should remind them that it took 4 years of painstaking work to arrive at an answer on just one specific odor problem. In the odor research field the terms quick and practical have no place in the same sentence.

It was gratifying to see that this program confirmed the desirability of operating with a dry coil. Preliminary research by this discussor 16 years ago indicated the

† Research Psychologist. Vision Research Laboratories, University of Michigan.

desirability of dry coil operation and 15 years of experience with 10's of thousands of individual dry coil room units has confirmed both that early research and the findings reported now on a much more factual basis by the present authors.

In the opening paragraphs the authors referred to the term *Air Conditioning Odor* as being specifically confined to coil odor of an unusual and infrequent nature associated almost solely with the specific material surfaces of the coil. I believe it is safe to say however that the term *Air Conditioning Odor* is more generally used by the public in referring to the deposition of odor molecules which became absorbed on the interior surfaces of the ducts, plenum chambers and particularly the dehumidifying coil surfaces. In other words I am trying to say that there actually is a distinction between *air conditioning odor* on the one hand and *coil odor or air conditioning coil odor* on the other hand.

The authors have done an outstanding job on the latter subject. If these talented authors could now find a basis for applying themselves to the other phase of the subject—air conditioning odor—I am sure that our industry would benefit greatly.

R. S. DILL, (Washington, D. C.): To get it in the record, I would like to ask Mr. Hubbard if he can tell us anything about the effects of depositions of dust or lint on the coils in influencing this odor generation problem and if there is any decomposition of material deposited on the coil that might contribute to the odor.

E. R. KAISER, (Cleveland, Ohio): At the Research Laboratory the Society is conducting a research program on odor measurement and control. The first phase of the work, which is a study of the effect of humidity and temperature on the perception of odor, is nearing completion. It was found that absolute humidity has an important effect on odor perception, an increase in humidity generally causing a decrease in the intensity of the odor, despite the fact that the odor concentration actually remained constant.

In view of these findings, it would be important if the authors would comment on whether the air humidities were relatively constant during the notations of odor intensity from the coils.

A second item of considerable interest on which further elaboration would be appreciated is the theory of why a metal or a coating of 1 type performs differently from another. Is there a chemical reaction between the odorants and the metals or surfaces?

The paper is an important contribution and will undoubtedly have a marked effect on the materials used for air-conditioning coils. The paper lends encouragement to further studies on odor adsorption by carpeting, drapes, and other interior surfaces that apparently retain odors for long periods after exposure to heavy concentrations of tobacco-smoke odor.

G. L. TUVE, Cleveland, Ohio: Just 1 short question. Did you use smokers or non-smokers on your panel?

Is there any difference in the reactions reported by these 2 groups?

F. W. McKENNA, St. Louis, Mo.: The speaker said that he found 1 job in 10 that, in his personal opinion, was unsatisfactory. If possible, I would like to have him elaborate a little bit on that. Do you have any further opinion as to why that 1 job in 10 was bad?

A. B. NEWTON, Wichita, Kansas: I might just ask a question or two. One thing I was wondering about as Mr. Hubbard spoke, is whether or not he could correlate the results of his laboratory tests with these 20 percent of the jobs that seemed to be in trouble in the field, or was that another part of the program? I think it would be an interesting thing to know about. Another point I have wondered about, a number of years ago we attempted to do a little odor work and thought it was necessary to find some way of amplifying the odor level so that people could perceive them more accurately, and we tried the idea of absorbing them on some material and then driving

them off so they would be more concentrated. I wonder if anything like that has been tried in the authors' work.

**AUTHORS' CLOSURE** (A. B. Hubbard): Dr. Beck is to be thanked for his kind comments.

Mr. Meek was also very kind. He corroborates that dry coils have the least odor pick-up. Our findings apply to coils normally dehumidifying. We noted, however, that when dew-point was changed, all other things being equal, odor pick-up was greatly reduced when the coils were dry and was worst in between when coils were wet but not dripping. Our terms were borrowed from service men, but Mr. Meek is undoubtedly correct in distinguishing *air conditioning odor* as a broader term than *coil odor*.

Mr. Dill requests something for the record regarding dust and lint deposits on coils as they may influence odor. In all cases observed the coils were preceded by filters so that we did not find appreciable deposits. We initially expected slimy deposits and, it's a joke on us, armed ourselves with specimen bottles and swabs when we first went into the field. We succeeded in getting faint green marks on our swabs; but nothing to analyze. The appearance and odor of coils did not correlate. Some of the cleanest looking surfaces smelled the worst.

Mr. Kaiser mentions that odor perception depends on temperature and humidity. We were not aware of this fact at the time. However, conditions were in a narrow range for other reasons, hence odor perception must have been constant.

Mr. Kaiser wonders why different metals or coatings performed differently. We believe the known facts of adsorption are sufficient to explain the different patterns observed. It is not necessary to postulate chemical reactions between odorants and surface materials.

Mr. Kaiser made a point of adsorption by other surfaces in the air conditioned space. This is an important source of stale odors. A common example is after-the-party smell. The mechanism is something like this. Cotton fabrics pick up the characteristic odor of tobacco. So, while the party is going on, the drapes and other fabrics are busily picking up tobacco smell. But the cotton prefers water vapor. Therefore, as humidity comes up in the latter part of the night, cotton trades back the worst part of tobacco odor for water vapor. Hence the foul smell the morning after.

Professor Tuve voices the common doubt concerning smokers vs non-smokers as panel members. We used both and find no significant difference. Panel members are allowed to live normal lives; indeed it is important that they be at ease in all respects. It is a good precaution to refrain from smoking immediately before a determination; just as one should not use strong personal perfumes or eat spicy, odorous foods. There is no reason why a smoker should be any less keen a judge of odors than a non-smoker. In any event, panel management includes cross-correlations designed to eliminate wild performance, whatever the reason.

Mr. McKenna asked for elaboration on why 1 job in 10 is unsatisfactory. This estimate includes all causes: coil odor, inadequate ventilation, unusual odor sources and poor housekeeping. No breakdown has been attempted, but the result might divide causes about equally between the above reasons.

Mr. Newton asks whether laboratory results could be correlated with the observed conditions of the small percentage of jobs having coil odor. We can not predict accurately whether a given job will develop coil odor. It depends on a critical balance between opportunity to pick up odors and opportunity to unload them. We can only say that a unit operated continuously on cooling, with no off-cycle time with fan on, has a maximum chance to have coil odor.

Mr. Newton wondered about the need for amplifying odors for easier perception. This was not necessary in the coil odor study. I can emphasize the point by describing my introduction to the phenomenon. At a night club in Greenwich Village the field engineer stood me on a chair so that my face would be in front of a grille. When he turned on the air conditioner my knees literally buckled. Then, and later, we did not need amplification.



**1548**

## A RAPID GENERAL PURPOSE CENTRIFUGE SEDIMENTATION METHOD FOR MEASUREMENT OF SIZE DISTRIBUTION OF SMALL PARTICLES

### Part II—Procedures and Applications

By K. T. WHITBY\*, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with the Department of Mechanical Engineering of the University of Minnesota.

**P**ART I of this paper† described the basic principles and apparatus of a new, rapid, general purpose centrifuge-sedimentation method designed for the economical determination of particle-size distribution in the 0.05 to 100 micron size range.

Since sedimentation analysis of particle size distribution is an indirect measurement, a number of factors must be considered when interpreting the results. A recent symposium of the *British Institute of Physics* has discussed many of these factors comprehensively.<sup>1</sup> In addition to those factors common to all sedimentation methods, a number are peculiar to this centrifuge method. The data and discussion that follow illustrate some of the most important conclusions reached as a result of numerous comparisons with other methods and evaluations by the various investigators using this method. Because reproducibility is affected by the techniques used, certain important procedures are described in detail.

#### SELECTION OF THE DISPERSION SYSTEM

As many investigators have pointed out, adequate dispersion is perhaps the most important factor affecting the validity of sedimentation results. For this reason, the choice of liquids, of dispersing agents and the method of dispersion depend more on the overall quality of the dispersion than on the physical properties of the system. Workable dispersion systems as well as practical methods for selecting the best system are discussed.

\* Research Associate, Mechanical Engineering Department, University of Minnesota.

† A Rapid General Purpose Centrifuge Sedimentation Method for the Measurement of Size Distribution of Small Particles, Part I—Apparatus and Method, by K. T. Whitby (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 33).

<sup>1</sup> Exponent numerals refer to references.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

## SEDIMENTATION AND FEEDING LIQUIDS

The particles to be analyzed must be practically insoluble in both the feeding and sedimentation liquids. Solubility may be determined from handbooks if the material is known, or by shaking a small amount of the dust and the liquid being tested in a test tube. Occasionally a mixture of materials will be encountered in which at least one of the components will be soluble in all the liquids tried. For such cases judgment must be exercised in interpreting the resulting size analysis.

Usually the feeding liquid is made up by mixing a miscible liquid of lower density with the sedimentation liquid in such a proportion that the density difference between the feeding and sedimentation liquid is about 10 percent of the density difference between the particles and the sedimentation liquid. It has also been found that the adverse effects of density streaming can be decreased further if the feeding liquid has a 20 to 50 percent greater viscosity than the sedimentation liquid. This is especially helpful when analyzing powders of high density, having coarse, relatively narrow size distributions.

Some of the most useful sedimentation liquids are given in Table 1. Mixtures

TABLE 1—USEFUL SEDIMENTATION LIQUIDS AND DISPERSING AGENTS

SEDIMENTATION LIQUID	$\rho_0$ — GM/CC 80 F	$\gamma_0$ — C.P. 80 F	DISPERSING AND WETTING AGENTS <sup>b</sup>
Water.....	0.997	0.863	Household Detergents, A, B, C, D, E, Tetra Sodium Phosphate and many others.
Acetone.....			E
Benzene.....	0.872	0.582	D
Carbon Tetrachloride.....	1.583	0.883	D
35% M.O. <sup>a</sup> —65% Kerosene...	0.827	4.004	D
1/3 M.O.—2/3 Carbon Tet....	1.332	2.132	D
1/3 M.O.—2/3 Benzene.....	0.866	1.122	D
Isopropal Alcohol.....			Sodium Oleate

<sup>a</sup> U.S.P. Heavy Mineral Oil.

<sup>b</sup> A—Non ionic wetting agent; B—Polymerized salts of aryl alkyl sulfonic acids; C—Lignin sulfonic acid compounds; D—Sodium sulfonate of mineral oil; E—Dioctyl sodium sulfosuccinate.

of benzene, carbon tetrachloride, naphtha and U.S.P. mineral oil can be made up to have just about any density and viscosity desired. The viscosity characteristics of these mineral oil mixtures are determined almost completely by the low viscosity component. Thus the viscosity characteristics of succeeding batches mixed in the same proportion will be close enough to make repeated viscosity determinations unnecessary.

Some appropriate feeding and sedimentation liquids for specific materials are tabulated in Table 2.

## DISPERSING AND WETTING AGENTS

Table 2 shows that for most materials it is necessary to add either a dispersing agent, or a wetting agent and sometimes both, to get a satisfactory dispersion. In addition to the data in Table 2, recommendations for many other materials may be found in a book by Herdan<sup>2</sup> (p. 451). If a dispersion system must be found for a material not listed in the literature then the dispersion criteria given further on in this paper or those given by Herdan<sup>2</sup> can be used.

TABLE 2.—DISPERSION SYSTEMS FOR VARIOUS MATERIALS ANALYZED BY THE CENTRIFUGE METHOD

ITEM	MATERIAL	SEDIMENTATION LIQUID	FEEDING LIQUID <sup>a</sup>	DISPERSING AGENT <sup>d</sup>	CONCENTRATION OF DISPERSING AGENT <sup>a</sup>	METHOD OF DISPERSION <sup>b</sup>	QUALITY OF DISPERSION
1	Glass beads						
2	Quartz	2.65	Mixture of M.O. Bz. C.T. or Naphthalene of correct density and viscosity	D	0.05	Shaking or Micro-stirrer	Excellent
3	Silica Carbide	3.13					
4	Aluminum Oxide	3.89					
5	Limestone						
6	Manganese Dioxide	5.45	60 Naphthalene, 40 Bz	D	0.1	Blender	Good
7	Chrome Yellow	6.16	Kerosene	D	0.1	Blender	Excellent
8	Titanium Oxide	3.8	70 H <sub>2</sub> O; 30 Acetone	C	0.2	Blender	Excellent
9	Clay	2.7	70 H <sub>2</sub> O; 30 Acetone	C, A	0.2:0.05	Blender	Excellent
10	Phosphate Fertilizer	3.28	70 H <sub>2</sub> O; 30 Acetone	B	0.1	Blender	Excellent
11	Calcium Carbonate	2.70	70 H <sub>2</sub> O; 30 Acetone	C	0.1	Blender	Excellent
12	Zinc Stearate	1.16	50 Acetone; 50 Naphthalene	—	—	Blender	Good
13	Arizona Road Dust	2.69	Mixture of Bz, C.T. and Naphthalene	D	0.05	Blender	Excellent
14	Arizona Road Dust	2.69	70 H <sub>2</sub> O; 30 Acetone	C, A	0.2:0.05	Blender	Excellent
15	Air Filter test dust—mixture of Arizona Road dust, carbon and lint.	2.5	70 H <sub>2</sub> O; 30 Acetone	C, A	0.2:0.05	Blender	Excellent
16	Fly ash	2.61	70 H <sub>2</sub> O; 30 Acetone	C, A	0.2:0.05	Shaking	Excellent
17	Biological Material <sup>d</sup>	1.37	50 Bz; 50 Naphthalene	D	0.1	Blender	Excellent
18	Flour <sup>a</sup>	1.44	50 Bz; 50 Naphthalene	—	—	Shaking in chamber	Excellent
19	Powdered Milk	1.30	50 Bz; 50 Naphthalene	—	—	Malted milk mixer	Excellent
20	Corn dextrin	1.50	50 Bz; 50 Naphthalene	—	—	Shaking in chamber	Excellent
21	Silica	1.35	50 Bz; 50 Naphthalene	D	0.5	Malted milk mixer	Excellent
22	Alfalfa meal dust	1.35	50 Bz; 50 Naphthalene	D	—	High Blender	Excellent
23	Citric acid	1.54	50 Bz; 50 Naphthalene	—	—	Speed Blender	Excellent
24	Cellulose-paper fibers	1.52	50 Bz; 50 Naphthalene	D	0.1	Shaking	Excellent
25	Flax Sheaves	1.48	50 Bz; 50 Naphthalene	—	—	Shaking	Excellent
26	Ground beet pulp	1.41	50 Bz; 50 Naphthalene	—	—	Shaking	Excellent
27	Cocoa	1.45	50 Bz; 50 Naphthalene	—	—	Shaking	Excellent
28	Soy bean meal	1.40	50 Bz; 50 Naphthalene	—	—	Shaking	Excellent
29	Egg albumin	1.30	50 Bz; 50 Naphthalene	D	0.5	Shaking	Excellent
30	Aspirin Powder	1.41	Naphthalene	—	—	Blender	Good
31	Powdered sugar	1.59	50 Iso; 50 Naphthalene	sodium oleate sugar	Sat.	Malted Milk Mixer	Excellent

<sup>a</sup> Figures in this column are percentages.<sup>b</sup> In all cases the blender is the high speed type.<sup>c</sup> Abbreviations: C.T. (carbon tetrachloride), Bz (benzene); M.O. (mineral oil).<sup>d</sup> Biological material is dry viable.<sup>e</sup> The flour is wheat, rye or rice.<sup>f</sup> Agents are identified in Table 1.

## DISPERSION CRITERIA

A test tube can be used to obtain a fair indication as to whether a given dispersion system is satisfactory. If a small amount of powder, shaken in a test tube with the system being tested, forms a turbid suspension with no visible flocculation and then settles as a sharply defined spot in the middle of the rounded bottom of the test tube, this is an indication that wetting and dispersion are satisfactory. On the other hand, if the particles settling on the bottom form an irregular sediment layer, it is a sign that sufficient interaction of the particle has taken place to cause trouble during the size analysis.

Direct observation of the sedimenting particles in the centrifuge tube is also a fairly good criterion. No flocculation or sticking of particles to the tapered portion of the walls of the tube should be visible. Sometimes such sticking is caused by improper cleaning of the tubes after changing from a hydrophobic to a hydrophilic sedimentation liquid or vice versa. Thorough brushing with acetone before changing liquids will usually eliminate this source of trouble.

The final height of the sediment column should show no change as the centrifuge speed is increased after sedimentation is complete except for soft materials such as flour. Even for flour no appreciable change is noticeable until the speed is increased over 2000 rpm. That this should be a good criterion is reasonable. Since essentially one size particle is settling at one time in the capillary, the void space should be constant and a minimum if the particles do not interact as they come to rest. Any interaction will tend to cause the void space to be greater than the minimum and will permit reorientation of the particles as the centrifugal field is increased. Compression of the sediment column with increasing centrifuge speed is therefore an indication that the dispersion system is not completely satisfactory.

Replicate particle size analysis with different amounts of mechanical agitation and different amount of dispersing agents, should give the same size distribution if the dispersion is complete. This is a very severe criterion and may be difficult to use if the particles are themselves moderately strong aggregates of smaller particles.

## DISPERSION METHODS

The dispersion method used will depend on the material being analyzed. In some cases considerable experimentation may be required to establish the severity and length of mixing required to get the desired degree of dispersion. For all but a few situations one of the 4 methods discussed later has been satisfactory.

Easily dispersed materials such as flour, glass beads abrasives and other relatively coarser or hard materials can usually be dispersed by shaking in the dispersing chamber with the appropriate feeding liquid.

For more difficult-to-disperse materials, the micro-stirrer illustrated in Fig. 6, Part I may be used. Dispersion directly in the feeding chamber is especially convenient for routine analysis because it is then unnecessary to prepare and transfer a separate suspension.

Materials which have more than a few percent by weight of particles below 1 micron usually require more severe mixing. For these either a malted milk mixer or a high-speed blender may be used.

For such very fine materials as paint pigments and for some coarser materials that tend to form strong agglomerates, severe working in a viscous medium may be required. One satisfactory technique is to mix a paste of powder in a heavy

TABLE 3—REPRODUCIBILITY DATA FROM 43 RUNS ON A SINGLE SMALL SAMPLE OF GLASS BEADS<sup>a</sup>

% BY WT. LESS THAN	<i>d</i> MICRONS	$\sigma$ MICRONS	$\sigma$ %
95	35.2	0.9	2.6
50	24.2	0.4	1.6
5	16.3	0.8	4.9

<sup>a</sup> Sed. Liquid— $\frac{1}{4}$  Heavy Mineral Oil,  $\frac{1}{4}$  Carbon Tetrachloride, Graduated tubes.

mineral oil on a glass plate with a spatula. After several minutes of mixing a small amount of the paste is diluted with a miscible liquid of low viscosity and placed in the feeding chamber.

If there is any question as to the effect of the dispersion method on a given material, comparative size analyses using different dispersion techniques can be run. Data of this kind often shed light on the nature of the agglomeration and size distribution of the primary particles.

*Concentration of Particles in Feeding Liquid:* The concentration of particles must give the desired sediment height in the tubes. Considerable experimentation has shown that 10 to 15 mm is satisfactory for graduated tubes and 4 to 5 mm in ungraduated tubes. The 0.75 mm tubes are used for all powders except those having a density less than about 1.8. The best layer of feeding liquid can be formed if the feeding chamber is from one-half to three-fourths full. Since the reproducibility does depend somewhat on the thickness of the feeding layer, care should be taken to use exactly the same volume of feeding liquid if highest reproducibility is required.

*Reproducibility of the Method:* Experience has shown that for the majority of applications, the reproducibility of a size analysis method is considerably more important than the accuracy defined by comparison with some other method.

TABLE 4—PARTICLE SIZE ANALYSES OF CALCIUM CARBONATE

MICRONS	% BY WEIGHT LESS THAN SIZE				AVG
	RUN 1	RUN 2	RUN 3	RUN 4	
9	96.5	97.8	97.0	97.9	97.3
6	83.9	85.7	84.0	86.3	85.0
4	65.5	65.9	62.0	66.3	64.9
2	33.3	30.7	30.0	31.6	31.4
1	11.5	10.9	11.0	11.6	11.2
0.7	6.9	5.5	5.0	5.3	5.7
0.4	2.3	2.2	2.0	2.2	2.2
0.2	0	0	0	0	0

PERCENT BY WEIGHT LESS THAN	<i>d</i> MICRONS	$\sigma$ MICRONS	$\sigma$ %
95	8.0	0.3	3.7
50	3.1	0.1	3.2
5	0.64	0.01	1.6

Sed. Liquid—Water; Graduated Tubes; Dispersed in high speed Blender.

TABLE 5—PARTICLE SIZE ANALYSES OF A HIGH SPEED MIXER BLENDED CAKE FLOUR

OPERATOR A, MARCH 6, 1953				OPERATOR B, AUGUST 4, 1953				
SIZE MICRONS	PERCENT LESS THAN SIZE			SIZE MICRONS	PERCENT LESS THAN SIZE			AVG
	RUN 1	RUN 2	AVG		RUN 1	RUN 2	RUN 3	
87.5	100.0	100.0	100.0	80	99.2	100.0	99.3	99.5
75.0	98.3	98.3	98.3	70	97.7	97.8	97.8	97.8
62.5	94.3	93.4	93.9	60	92.4	92.1	92.4	92.4
50.0	82.0	82.0	82.4	50	84.8	82.8	85.5	84.4
37.5	69.8	67.9	68.9	40	74.9	72.8	76.0	74.4
25.0	49.4	48.4	48.9	35	68.1	65.6	69.5	67.7
12.5	19.3	18.0	18.6	30	60.2	59.2	62.9	60.8
Sed. Liquid—Benzene Graduated tubes				25	49.8	49.1	51.4	50.8
				20	34.6	33.4	37.5	35.2
				12	15.6	15.4	15.0	15.3
				8	8.6	8.9	8.4	8.6
				5	4.2	3.2	3.4	3.6
				2.5	1.1	1.0	1.1	1.1
				1	0	0	0	0

This is true because most methods have certain inherent biases which are usually dependent on particle shape and other variables.

Tables 3, 4, and 5, show typical reproducibility data for three different situations. Table 3 is a summary of 43 replicate runs on a single small sample of relatively narrow size range glass beads made as part of an evaluation of the method. The particle sizes at the 95, 50, and 5 percent less than points on the distribution, were read from the plotted and smoothed cumulative size distribution curves. These data are indicative of what may be expected on easily dispersed mineral materials having fairly narrow size distribution.

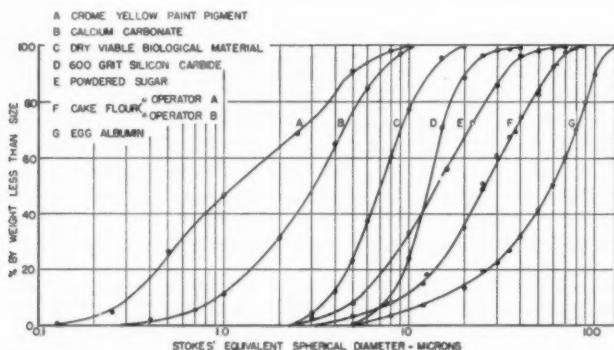


FIG. 1. TYPICAL PARTICLE SIZE ANALYSES ON DIFFERENT MATERIALS

The 4 replicate runs on calcium carbonate shown in Table 4 are typical of what may be expected on fine powders. The average of these 4 runs is shown in Fig. 1, Curve B.

Table 5, and Curve F of Fig. 1, show replicate runs on a sample of cake flour made by different operators using different reading schedules. This level of reproducibility was attained only after passing the flour through a high-speed mixer.

*Comparisons with Other Methods:* For those applications where it is desired to further evaluate the meaning of the Stokes equivalent spherical diameter, it is necessary to make comparisons with other methods. Differences between sedimentation and other particle size methods such as the microscope arise from 2 different sources. First, the sedimentation size depends on particle drag which depends on particle shape and particle roughness. This is a fundamental characteristic common to all sedimentation methods. Second, most sedimentation methods have certain other biases determined by the particular apparatus and sensing method used. While it is desirable to minimize the latter biases, compromise is usually necessary to achieve the desired economy of operation. Though these biases are difficult to evaluate separately since they depend on the material being analyzed and other variables that depend on the particular situation, a qualitative discussion is helpful in judging the possible effect of each. Following are a number of these factors.

1. *Conversion of Volume Measurement to Weight Measurement:* Actually this method measures the volume-size distribution. However, if the particle shape is constant with size then the void fraction will be the same for the different sizes because only one size particle is coming to rest in the capillary at any given time. Under these conditions the assumption of a linear relationship between the sediment height and the weight-distribution is satisfactory.

2. *Decreasing Sedimentation Height with Increasing Sediment Height:* The magnitude of this effect depends on the ratio of the total sediment height to the assumed settling distance. For the graduated tubes the maximum sediment height is about 15 mm and the assumed settling distance is 100 mm. Under these conditions the theoretical maximum error is about 4 percent. Though the net effect of this error would be to narrow the actual size distribution, in practice factors 3 and 4 following, tend to cancel it.

3. *Density Streaming:* A suspension of particles in a liquid tends to act like a liquid having a density greater than that of the pure liquid. If such a suspension is then floated on the surface of the pure liquid, irregular streamers of the suspension will be observed settling through the liquid. Such streamers are minimized in this method by suspending the particles in a feeding liquid of lower density and higher viscosity than the sedimentation liquid and by using low concentrations of particles. It has been discovered, however, that this streaming, though minimized, tends to spread very narrow size distributions slightly making them appear broader than they actually are.

This effect is illustrated in Fig. 2, where size distribution measurements by microscope, by standard Andreason pipette and by centrifuge sedimentation, on a narrow size fraction of glass beads are compared. Though the medians of the distributions measured by the different methods are nearly the same, the geometric standard deviation by the centrifuge method is greater than by microscope. This effect is serious only for distributions having a geometric standard deviation less than about 1.2 except for materials having a density above 4. For these high density materials some spreading may occur at geometric standard deviations up to 1.5, if the median is above  $20 \mu$  (microns).

Fig. 3 (Part I) illustrates the relatively good agreement with the Andreason pipette for broad distributions.

4. *Hindered Settling in the Tapered Portion of the Tube and in the Capillary:* Since the tapered portion of the tube and the capillary comprise about 50 percent of the total

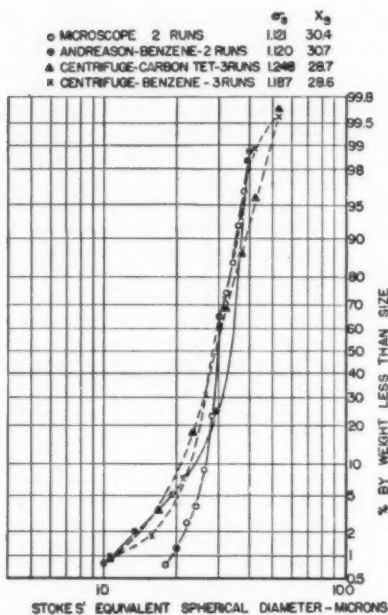


FIG. 2. COMPARATIVE SIZE ANALYSES ON A ROLLER TYPE ELUTRIATOR CLASSIFIED SAMPLE OF GLASS BEADS ( $\rho = 2.34$ )

settling height, it might be imagined that hindered settling might cause considerable difficulty.

Actually it has been observed that if there is no sticking of particles on the walls of the tapered portion and no choking at the entrance of the capillary caused by all the particles of a narrow distribution arriving at once, then these effects are small. Sticking can be eliminated by proper dispersion systems and proper cleaning, while choking is eliminated by using the projection system to permit accurate reading of small sediment heights.

Experience has shown that the comparative accuracy and reproducibility is satisfactory for the measurement of all but very narrow size distributions. Even for this special case, the method is useful if it is evaluated against the microscope for the particular material and conditions of interest. The following study of size analysis results on wheat flour serves to illustrate how such an evaluation may be carried out.

The object of this evaluation was to obtain data by means of which the Stokes equivalent spherical diameter within the range of flour particle sizes could be converted to microscope, sieve, or count and weight size. Because of its low viscosity benzene is usually used as the sedimentation liquid for routine size analysis of wheat flour. Unfortunately the largest flour particles are above the size where Stokes' law begins to be in error. Thus the lack of agreement between the centrifuge and

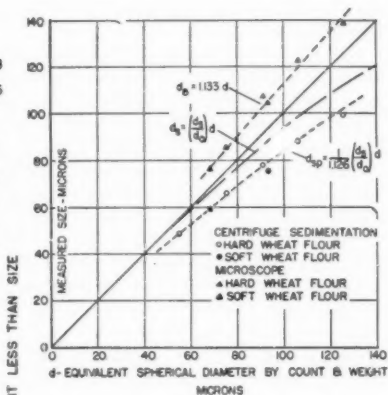


FIG. 3. COMPARISON OF THE MASS MEDIAN OF FLOUR FRACTIONS MEASURED BY DIFFERENT METHODS WITH THE CALCULATED DEVIATIONS FROM STOKES' LAW

other methods is a combination of deviation from Stokes' law, particle shape and the other effects peculiar to the centrifuge method.

Deviation from Stokes' law begins when the terminal velocity of a particle exceeds a certain Reynolds number and the flow around the particle begins to change from streamline to turbulent. Oseen has derived an equation from which the error in Stokes' law can be calculated as a function of particle size. Oseen's equation is given by Rose<sup>3</sup> as:

$$d_o = d_s (1 + 0.095 R_s) \quad (1)$$

where

$d_o$  = the true diameter of a spherical particle.

$d_s$  = the diameter of a spherical particle calculated from Stokes' law for a given terminal velocity.

$R_s$  = the Reynolds number of the sedimenting particle.

If  $R_s$  and  $d_s$  are replaced by their equivalents in terms of the constants of the particular system under consideration, then Equation 1 becomes:

$$d_o/d_s = 1 + (0.095 \rho_o d_s h / \eta_o t) \quad (2)$$

From Equation 2, the ratio  $d_o/d_s$  can be calculated for any given values of  $h$ ,  $t$ , and  $d_s$ . This ratio is tabulated in column 9, of Table 6, as a function of particle size. Note that it ranges from about 1 at 50  $\mu$  to 1.264 at a true particle size of 177  $\mu$ . This ratio between true particle and calculated particle size applies only

TABLE 6—RELATIONSHIP OF SEDIMENTATION SIZE TO SIZE MEASURED BY MICROSCOPE, COUNT AND WEIGHT AND SIEVING FOR WHEAT FLOUR<sup>a</sup>

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
$d_{sp}$	$d_o$	$d$	$d_b$	$d_n$	$\frac{d}{d_{sp}}$	$\frac{d_b}{d_{sp}}$	$\frac{d_n}{d_{sp}}$	$\frac{d_o}{d_s}$
140	177.0	199.2	225.3	217.2	1.423	1.610	1.552	1.264
120	138.7	156.0	176.0	169.9	1.300	1.469	1.417	1.156
100	108.5	122.0	177.8	133.0	1.220	1.378	1.330	1.085
80	83.5	94.1	106.2	102.5	1.176	1.328	1.281	1.043
70	72.0	81.0	91.5	88.2	1.157	1.307	1.260	1.028
60	61.2	68.5	78.0	75.2	1.148	1.298	1.252	1.020
50	50.5	56.8	64.2	62.0	1.137	1.283	1.240	1.010
40	40.0	45.0	50.8	49.0	1.126	1.271	1.227	1.000
30	30.0	33.8	38.1	36.8	1.126	1.271	1.227	1.000
20	20.0	22.5	25.4	24.5	1.126	1.271	1.227	1.000

<sup>a</sup> All dimensions in microns.

to spherical particles. For irregular particles an additional factor which includes the effects of shape and the biases of the sedimentation method must be determined by comparison of the sedimentation analyses with analyses by the other methods. Because it is difficult to obtain accurate microscope size distributions on the relatively broad original distributions a series of narrow sieve fractions of the flour were prepared by techniques described elsewhere<sup>4</sup>. The particle size distributions of these narrow fractions were then measured by centrifuge sedimentation, by microscope and by a count and weight technique that yields the average particle volume of the size fraction<sup>4</sup>. If the particle shape is assumed to be spherical, a

count and weight equivalent spherical diameter can be calculated. This diameter designated by  $d$  is given in column 2, Table 7. With reasonable care  $d$  can be reproduced within  $\pm 0.5$  percent for fractions having a mass median above  $30 \mu$ . Therefore  $d$  is a good reference to which the size measured by other methods can be compared.

Table 7 gives the mass median particle size as measured by count and weight, centrifuge, microscope and the mean nominal mesh opening of the 2 sieves used to make each fraction. In Fig. 3, the microscope and sedimentation size are

TABLE 7—PARTICLE SIZE DATA ON WHEAT FLOUR SIEVE FRACTIONS

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
FRACTION	$d$	$d_b^a$	$d_a^b$	$d_{sp}$	$d_o$	$\frac{d_s}{d_{sp}}$	$\frac{d_a}{d}$	$\frac{d_b}{d}$
Hard Wheat								
100/120	125.5	138.7	137.0	98.3	113.2	1.152	1.074	1.087
120/140	106.0	122.5	115.0	88.0	98.0	1.112	1.086	1.156
140/170	91.0	107.0	96.5	78.0	86.3	1.107	1.061	1.176
170/230	75.3	85.4	75.0	66.2	72.7	1.100	0.977	1.134
230/325	55.0			48.7	54.5	1.120		
Soft Wheat								
120/170	93.1	104.4	106.5	75.7	88.0	1.162	1.143	1.122
170/230	68.2	76.6	75.0	59.5	66.5	1.117	1.100	1.123
Mean value						1.126	1.074	1.133

<sup>a</sup> Dimension measured is breadth of particle in column 2.

$d_a$  = mean of nominal sieve mesh opening of two sieves used to make the fraction.

plotted on the ordinate versus the count and weight diameter  $d$  on the abscissa. Also  $d_o$  is plotted on the abscissa versus  $d_a$  on the ordinate to indicate the deviation of the sedimentation results as calculated from Oseen's law.

Note that the equivalent spherical diameter by sedimentation,  $d_{sp}$ , is less than the count and weight spherical diameter  $d$  and less than the diameter calculated from Oseen's law by approximately a constant factor. This factor tabulated in column 7 of Table 7, includes the effects of particle shape and the biases of the centrifuge method relative to the count and weight method. This factor, represented by the ratio  $d_s/d_{sp}$ , and the one resulting from the deviations from Stokes' law may be summed up in an equation of the form:

$$d = d_o = (d_s/d_{sp}) (d_o/d_a) d_{sp} \dots \dots \dots (3)$$

where

$d_{sp}$  = Stokes' equivalent spherical diameter for the irregular particles calculated from the observed terminal velocity.

$d$  = the equivalent spherical diameter by count and weight.

$d_o$  = Oseen's spherical diameter—true diameter of a spherical particle.

$d_a$  = Stokes' equivalent spherical diameter calculated from the observed terminal velocity of a spherical particle.

For the data of column 7, Table 7 there is no significant variation in the ratio  $d_s/d_{sp}$  with particle size so that  $d_s/d_{sp}$  may be taken equal to the mean of the seven

values. Therefore, for flour in benzene:

$$d = 1.126 (d_0/d_s) d_{sp}$$

where

$d_0/d_s$  = a function of particle size calculable from Equation 2.

In a similar manner, the ratios  $d_a/d$  and  $d_b/d$ , were calculated and tabulated in Table 7. From the mean values of these ratios in combination with the ratio  $d_0/d_s$  calculated from Equation 2 the data of Table 6 were obtained. Using these data it is possible to convert  $d_{sp}$  to the corresponding count and weight, microscope or nominal sieve size.

#### APPLICATIONS OF THE METHOD

Table 2 illustrates the great diversity of materials and conditions under which the method has been used. Not included in the table are a great many industrial dusts of unknown or uncertain chemical composition which have been analyzed. For such dusts the exact meaning of the particle size calculated from Stokes' law depends on how accurately the density is known. If sufficient powder is available for a conventional density determination, a particle size based on the average density can be calculated. The error in the particle size will be approximately equal to:

$$\text{percent error in } d = 100 \left[ 1 - \sqrt{\frac{\rho_{\text{assumed}} - \rho_0}{\rho_{\text{true}} - \rho_0}} \right]$$

The seriousness of this error depends on the application of the size analysis. Few applications have arisen in which this error has been a serious factor.

Fig. 1 illustrates a number of size analyses on different materials having different size distributions. For low density materials, such as flour, egg albumen and powdered milk the method is useful up to a maximum particle size of about  $175 \mu$  if the deviations from Stokes' law are taken into account. Curve C is interesting because the entire distribution was determined by centrifuge without any gravity settling. It also shows that up to 12 centrifuge times may be used if necessary.

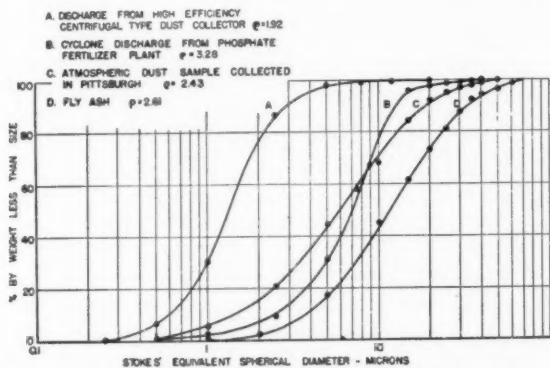


FIG. 4. PARTICLE SIZE ANALYSES ON SEVERAL TYPICAL DUSTS

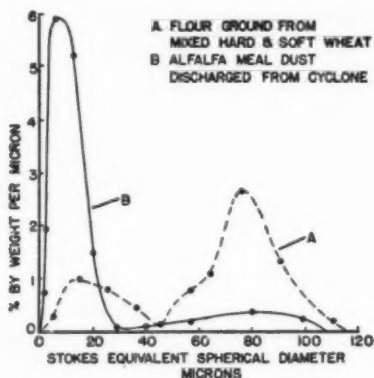


FIG. 5. TWO MATERIALS SHOWING A BIMODAL DISTRIBUTION. FOR THE FLOUR THE PEAK AT 17 MICRONS IS CAUSED BY FREE STARCH GRANULES FROM THE SOFT WHEAT AND THE ONE AT 75 MICRONS BY HALVES OF WHOLE CELLS FROM THE HARD WHEAT

The distributions of Fig. 4 are typical of a great variety of industrial dusts that have been analyzed. Curve B shows the abrupt break at about  $14 \mu$ , which is typical of the average industrial cyclone.

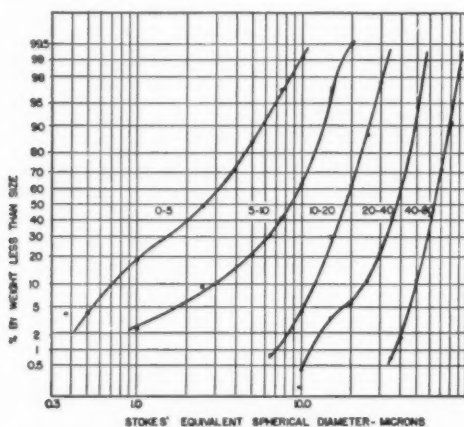


FIG. 6. SIZE DISTRIBUTION OF ROLLER TYPE ELUTRIATOR CLASSIFIED FRACTIONS OF A.C. FINE AIR CLEANER TEST DUST

Curve D is an analysis of a fly ash sample collected by electrostatic separator at the Paddy's Run Plant of the Louisville Gas and Electric Co.

The atmospheric dust sample curve C is typical of a number of such samples taken from electrostatic air cleaners in Pittsburgh. The application of the centrifuge method to the size analysis of atmospheric dusts will be discussed in a future paper.

Work is being done to extend the method to milligram quantities of atmospheric dusts collected by filter, by electrostatic sampler or other methods. At present the smallest sample which can be analyzed is about 1 mg, but it is hoped that this can eventually be reduced to about 0.2 mg by using finer capillaries in the tubes.

Two materials having bimodal size distribution are shown in Fig. 5. This method is especially valuable for studying such distributions because of the ease with which extra points on the distribution curve can be determined.

Figs. 6 and 7 show size analysis on dust fractions classified by two different methods from A.C. Fine Composite Test Dust. From this type of log-probability plot the goodness of the classification is easily seen. It will be noted that even though some of these were reclassified many times, there is still considerable material below the designated lower limit of the fraction. This characteristic usually is caused by the inability of the classifier to separate the fine particles adhering to the large.

The centrifuge method has also been useful in the development of fine grinding and classifying equipment. A few points on the distribution curve of the ground product could be determined rapidly enough so that adjustments and changes could be made without shutting the machine down. This speed in analyzing results permitted considerable economy of material and time. Likewise in several other applications the economical determination of large numbers of size distributions has permitted statistical studies that would have otherwise been impractical.

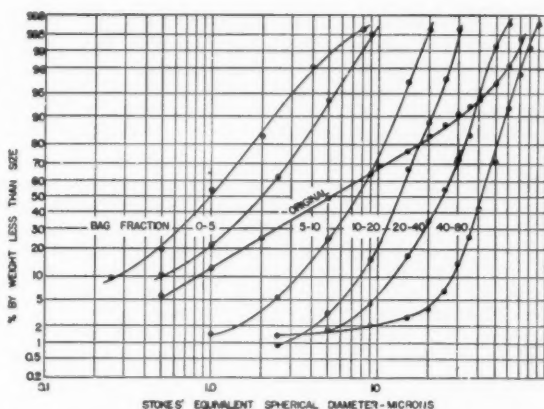


FIG. 7. SIZE DISTRIBUTION OF FRACTIONS OF A.C. FINE AIR CLEANER TEST DUST CLASSIFIED BY REPEATED RUNS THROUGH A COMMERCIAL CYCLONE TYPE AIR CLASSIFIER

## ACKNOWLEDGMENTS

The author is grateful for the help from the following individuals during the development of this method: Dr. R. C. Jordan, head, Mechanical Engineering Department, University of Minnesota, for making laboratory facilities available, and Prof. A. B. Algren, head, Heating, Air-Conditioning and Refrigeration Division, Mechanical Engineering Department, for activating the research project.

The author gratefully acknowledges the courtesy of the following organizations in providing certain samples and data presented in this paper: Minnesota Mining and Manufacturing Co. for providing the data given in Tables 3 and 4, and to General Mills, Inc. for the data of Curve C, Fig. 1; Superior Grain Separator Co., American Air Filter Co., Farr Co., *Industrial Hygiene Foundation*, Pillsbury Mills and the Fram Corp. for providing certain samples of dust analyzed and used as illustrations.

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2. *Small Particle Statistics*, by G. Herdan (Elsevier Publishing Co., New York 1953).
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## DISCUSSION

R. L. KUEHNER, York, Penna., (WRITTEN): In the past our industry has depended on repeated air passage through low efficiency filters for particulate air sanitation; screens to keep the birds off our evaporator coils. Further we have long labored under the impression that outside air is fresh air. The acute stage of air pollution shows that particulate matter in air has insidious hazards greater than just making things dirty. Our standards of air cleanliness are necessarily changing. To make economic progress toward air sanitation standardization we must first identify and set desirable limits on the air contaminants in question. We must then rate the efficiency of control measures. Since both these phases involve particle size distribution, the first logical step is practical measuring methods of the nature described in Dr. Whitby's paper. I believe the results in this and his preceding paper show real and valuable progress in this problem.

AUTHOR'S CLOSURE: Dr. Kuehner's recognition of the importance of the measurement of particle size distribution to the understanding of the air cleaning problem is appreciated. Although measurement of size distribution is usually only one of a variety of evaluations that must be made, it is the basis for analysis of air cleaner efficiency as a function of particle size. Good particle size versus efficiency evaluations over the range from 0.3 to 60 microns could eliminate much of the present confusion surrounding the performance of air cleaners on different dusts whether atmospheric or artificial.

Since the publication of the papers on the method, numerous inquiries have been received about the commercial availability of the necessary special apparatus. One commercial organization is undertaking the development of the apparatus.



**1549**

## SIZE DISTRIBUTION AND CONCENTRATION OF AIR-BORNE DUST

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**This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., in cooperation with the Mechanical Engineering Department, University of Minnesota, Minneapolis, Minn.**

SINCE AIR CLEANING devices are used under widely varying conditions of dust loading and dust character, it has become increasingly desirable to have available dust sampling and evaluation methods that can be used to determine the dust problem under given conditions. Such methods must be both comprehensive and economically practical. The work reported here describes a combination of methods that appears to meet this need for a practical method of obtaining air-borne dust samples and evaluating their concentration, character and particle size distribution for engineering purposes.

### SAMPLING METHODS

The desirability of making a size distribution analysis on the collected samples of air-borne dust limits the sampling methods that can be used.

Thermal precipitation is impractical because of the extremely low sampling rates permissible.

Liquid impingers are not efficient collectors of particles below about  $1\ \mu$ . Jet impactors<sup>1</sup> require careful design and use to collect small particles in a manner suitable for accurate size analysis of the particles below  $1\ \mu$ .

Of the remaining methods, sampling by filter and by electric precipitator appeared to be the most adaptable to the needs of this study.

Electric precipitation permits reasonably high sampling rates and easy separation of the sample from the collection chamber.

Of the high efficiency filter media suitable for aerosol sampling<sup>2</sup>, only the millipore filter was found suitable for the collection of samples for size analysis. These filters are efficient enough so that they may be considered absolute on most air-borne dusts<sup>2,3</sup>. Microscope size analysis, up to the limit of resolution of the light

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<sup>1</sup> Exponent numerals refer to References.

<sup>2</sup> Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

microscope, can be made on the filter directly. The solubility of these filters in acetone and certain other organic solvents permits ready dispersion of the collected particles into a suspension suitable for sedimentation size analysis.

#### MEASUREMENT OF PARTICLE SIZE DISTRIBUTION

Problems involved in the measurement of the size distribution of air-borne dusts are more difficult than might appear at first thought. Air-borne dust usually has a broad size distribution ranging from above  $30\ \mu$  to below  $0.3\ \mu$ . This broad distribution severely limits the usefulness of light microscopic methods because of the limit of resolution at about  $0.3\ \mu$  and because of the small probability of seeing the large particles.

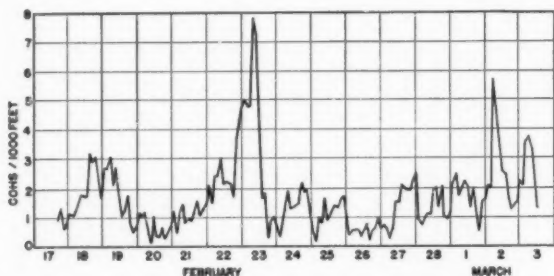


FIG. 1. RELATIVE DUST CONCENTRATION MEASURED IN THE UNIVERSITY OF MINNESOTA PARTICLE TECHNOLOGY LABORATORY FOR THE PERIOD COVERED BY THIS STUDY AS MEASURED BY THE STANDARD AISI SAMPLER

Sedimentation methods also suffer from certain limitations. Gravity sedimentation, whether in air or liquid, is time-consuming and unreliable below about  $1\ \mu$ . Liquid sedimentation methods, using centrifuges to increase the speed and convenience of analysis<sup>4</sup>, measure either the volume or weight distribution. The weight or volume readings are least accurate at the fine end of the distribution where the accuracy must be the highest if the weight distribution is to be transformed to a surface or number distribution. Another very real problem in using liquid sedimentation methods is the determination of the exact relationship between the measured distribution and the actual distribution of aggregates in the air.

In view of these serious limitations of any one method of size measurement, it is probable that any procedure for measuring the size distribution of air-borne dusts should incorporate independent measurements by more than one method, if confidence is to be placed in the results.

#### APPARATUS AND METHODS

As a result of preliminary experimental work, microscope size measurement on millipore filters and centrifuge sedimentation on samples collected both wet and dry with the electrostatic sampler were selected for further study. During the course of the work, successful procedures were developed for collecting samples on the

millipore filters for sedimentation. Sampling with the electrostatic sampler was, therefore, discontinued in favor of sampling with the millipore filters.

To provide a completely independent check on these 2 methods, the optical density of simultaneous samples collected on chemical filter paper, glass filter paper and millipore filters was compared with the equivalent optical density calculated by light scattering theory from the size distributions.

All samples taken were of the normal laboratory air existing at the time. As shown in Fig. 1, the dust spot discoloration measured by the standard *AISI* sampler<sup>6</sup> ranged from relatively clean to moderately dirty.

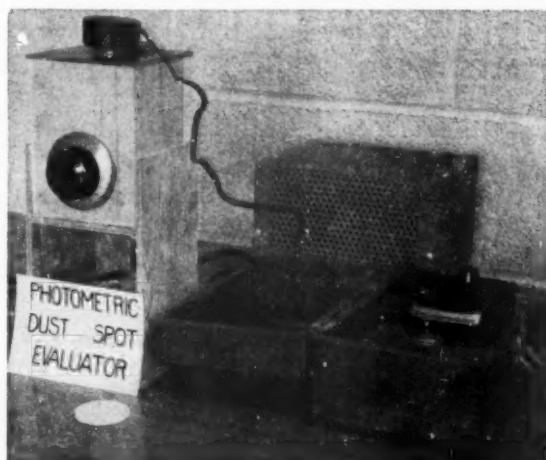


FIG. 2. PHOTOMETRIC SPOT EVALUATOR

#### SIZE ANALYSIS BY MICROSCOPE

Hydrosol type millipore filters were used for sampling because of their lack of grain when viewed under a high power oil immersion lens. Sampling time and rate were adjusted to give a final optical density of about 0.03. After first measuring the optical density of the filter with the photometer shown in Figs. 2 and 3, the microscope slide was prepared as follows:—a  $\frac{1}{2}$ -in. sq. of the filter was placed face down on a cover slip and, after wetting with immersion oil, the cover slip was carefully placed on a slide, care being taken to prevent any relative motion between the filter and the cover slip.

Counting was done with a Porton reticule in a microscope equipped with a 10X ocular, a 97X 1.25 numerical aperture objective and a blue diffusing filter below the substage condenser. A multistage count method, suggested by Work<sup>6</sup>, was used because of the wide size range of the dust. Approximately 1000 particles in at least 50 fields were counted. The results from a typical count are shown in columns 1 and 2 of Table 1, and also plotted in Fig. 4, along with the equivalent volume distribution calculated with and without the single  $20\ \mu$  particle.

## SIZE ANALYSIS BY SEDIMENTATION

The electrostatic sampler used is shown in Fig. 5. The chamber is rolled in at each end so that if desired about 15 ml of a liquid can be placed inside for wet collection. Rotation of the tube at 10 rpm about its axis maintains a thin film of liquid on the inside of the tube. A 100 mesh stainless steel screen is placed over the tube entrance to keep out large particles which would interfere with the sedimentation analysis. A separate evaluation method for these large particles is under development.

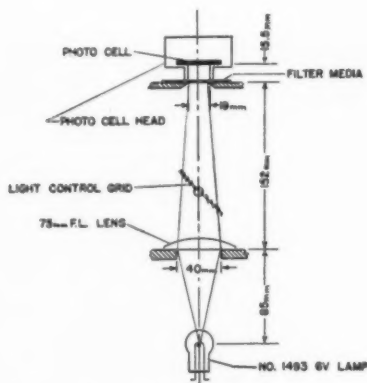


FIG. 3. OPTICAL SYSTEM OF DUST SPOT EVALUATOR

USP light mineral oil was used when the sampler was operated wet. At the end of a run, the mineral oil was washed out with acetone to which just enough naphtha had been added so that all of the oil would dissolve. The suspension was then centrifuged in a small laboratory centrifuge for about 1 hr to precipitate all particles above about  $0.15 \mu$ . The supernatant liquid was then siphoned off and the particles resuspended in sufficient 30 percent naphtha, 70 percent acetone mixture to fill the feeding chamber of the centrifuge tube. Particle size distribution was then determined using the apparatus and procedures previously described<sup>4</sup> with acetone as the sedimentation liquid. Acetone is the only sedimentation liquid tried which yielded a size distribution which would check with the microscope and photometric data. This is fortunate because the low viscosity of acetone (0.3 centipoise) permits relatively rapid size determinations to below  $0.2 \mu$ .

When the electrostatic sampler was operated dry, the precipitated dust was washed from the tube with pure acetone. The rest of the procedure was identical to that just described.

Samples collected on the aerosol type millipore filters were released by dissolving the dirty filter in pure acetone in a centrifuge tube and then following the rest of the procedure.

All of the sedimentation size analysis work here reported was done with 0.75 mm bore capillary tubes and the apparatus previously described<sup>4</sup>. In order to

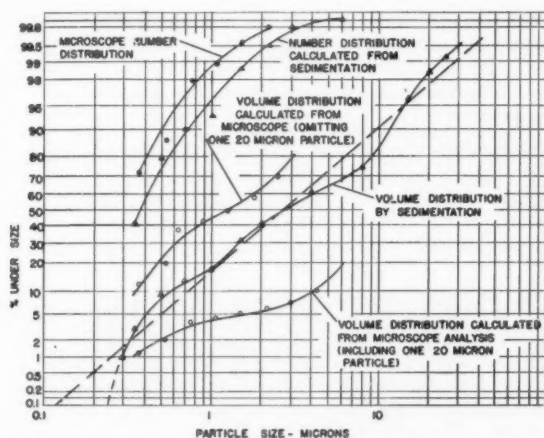


FIG. 4. COMPARISON OF MICROSCOPIC AND SEDIMENTATION SIZE ANALYSIS ON DUST SAMPLE COLLECTED FEBRUARY 23

get sufficient sediment height, the sampler was run 4 to 6 hr at a flow rate of about 2 cfm when sampling the relatively clean air in the laboratory. It is believed that special tubes and slight modification of the optical system of the reading devices will eventually permit a sufficient sample to be obtained in 1 to 2 hr and eliminate the necessity of taking a separate sample for photometric evaluation. The amount of sample required at present gives too high an optical density for accurate photometric evaluation.

TABLE 1—CALCULATION OF EFFECTIVE SCATTERING AREA FOR ATMOSPHERIC DUST SIZED BY MICROSCOPE

(1)	(2)	(3)	(4)	(5)	(6)
$d$ MICRONS	$n^a$	$d^3$ (MICRONS) <sup>3</sup>	$nd^3$	$K$	$Knd^3$
<0.447 <sup>b</sup>	4100	0.090	369	1.30	480
0.447	804	0.199	160	2.96	474
0.634	664	0.402	267	4.30	1150
0.895	75	0.80	60	3.56	214
1.268	25	1.61	40	2.50	100
1.82	14	3.31	46	2.15	99
2.51	7.1	6.30	45	2.00	90
3.58	6.1	12.8	78	2.00	156
5.06		25.6		2.00	
7.15		51.0		2.00	
10.13		102		2.00	
14.3		204		2.00	
20.2	1	408	408	2.00	816
					3579 <sup>c</sup>

<sup>a</sup> = for 82 fields; <sup>b</sup> = assumed size = 0.3 <sup>c</sup> Figure is  $\Sigma K_1 n_1 d^3$ , in (microns)<sup>3</sup>. (Sample obtained 2/23/55).

Some typical size distributions of air-borne dusts determined in this way are given in Fig. 6.

#### PHOTOMETRIC MEASUREMENTS ON DUST SPOTS

Though photometric measurements of dust spots on filter paper can not be made to yield a complete size distribution, such measurements are related to the total projected area of the particles in a given quantity of air. Therefore, these measurements offer an independent method of checking the size distribution and concentration measurements made by other methods. Derivation of the relationships between photometric data and the size distributions is given in the Appendix.

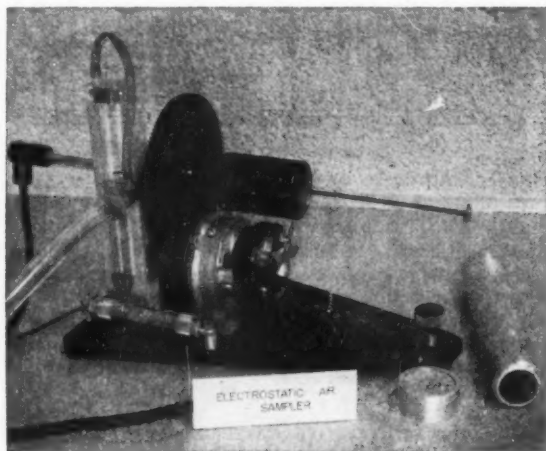


FIG. 5. ELECTROSTATIC AIR SAMPLER HEAD SHOWING ELECTRODES, SAMPLE TUBE, AND LINT SCREEN

Two different methods and 3 different filter media were used for dust spot sampling. For a continuous 24-hr record, an *AISI* automatic air sampler<sup>6</sup> was used. This sampler automatically takes a sample every 2 hr on No. 4 filter paper tape. One of the limitations of this instrument is the necessity of comparing light transmission adjacent to the spot with that through the spot in order to calculate the relative concentration in  $\text{colst per } 1000 \text{ ft}$  as defined by Equation A-1 (see Appendix). For the No. 4 filter paper and for some glass fiber that was also tried, the standard deviation in light transmission between adjacent areas 1.5 in. apart averaged about 2 percent. It may be seen from Table 2 that the probable percent error in the calculated dust concentration becomes rather large at low dust levels.

When more accurate photometric data were desired, samples were taken on filter media held in a commercial holder for the 47 mm size millipore filters and the light transmission evaluated over the same area before and after sampling.

<sup>†</sup> A measuring unit, the amount of carbonaceous material corresponding to an optical density of 0.01; also optical density  $\times 100$ .

TABLE 2—PROBABLE ERROR IN DUST CONCENTRATION MEASURED BY THE *AISI* SAMPLER, CAUSED BY VARIATIONS IN LIGHT TRANSMISSION THROUGH THE FILTER TAPE

COHS PER 1000 FT	OPTICAL DENSITY	95% CONFIDENCE INTERVAL <sup>a</sup> ON COHS PER 1000 FT
4	0.190	±10%
2	0.950	±20%
1	0.470	±32%
0.5	0.024	±73%

<sup>a</sup> For 2 percent standard deviation in light transmission.

The photometer used to evaluate all of the dust spots is shown in Figs. 2 and 3. It consists of a high grade electronic voltage regulator supplying power to the photometer lamp, the photometer proper shown in Fig. 3, and a commercial microscope light meter. The latter is convenient because its high sensitivity permits full scale deflections for relatively opaque filter materials such as the millipore and glass fiber papers. Table 3 gives typical light transmission values for the various

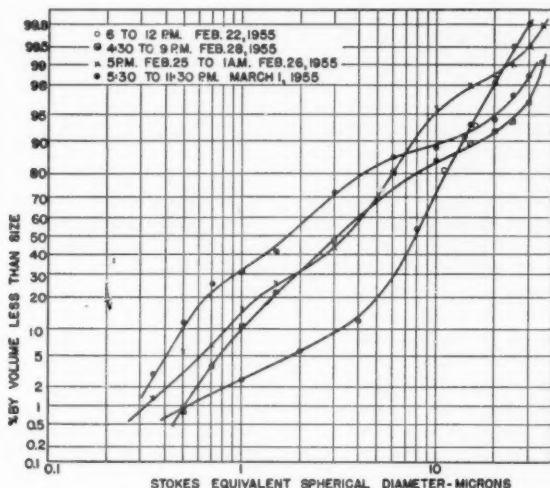


FIG. 6. TYPICAL SEDIMENTATION SIZE DISTRIBUTIONS OF ATMOSPHERIC DUST

filter materials. With reasonable care, light transmission measurements accurate to about 0.3 percent can be made with this apparatus. Though this instrument is reasonably satisfactory, a more versatile photometer of the compensated, null type, which should give 0.1 percent accuracy, is now being designed.

To determine the exact relationship between the optical density as measured on the filters and that predicted from light scattering theory, a number of additional

experiments were carried out with varying photocell arrangements and with the filter immersed in liquid. These experiments were performed using the commercial photometer and a standard microscope and lamp. By using various numerical aperture objectives on the microscope, light could be accepted through varying cone angles.

TABLE 3—PERCENTAGE LIGHT TRANSMISSION OF VARIOUS FILTER MEDIA

MEDIA	% TRANSMISSION
Opal glass standard.....	26.7
Glass fiber paper.....	4.0
Filter paper No. 41.....	16.3
Filter paper No. 4.....	15.0
Aerosol Type Millipore Filter.....	2.8
Hydrosol Type Millipore Filter.....	2.3

### RESULTS AND DISCUSSION

Table 4 summarizes the data obtained from the various size distribution measurements and calculations. Column 2 gives the particle densities that were assumed in calculating the centrifuge tables and for determining the dust concentrations in column 4. As a result of average density determinations on dust taken from a number of electrostatic air cleaners, it was originally believed that the average density of atmospheric dust was about 2.1 g per cc. However, some approximate specific gravity separations indicated that density of the dust samples being collected in the laboratory was probably closer to 1.6. This value was, therefore, used for some of the later runs. Work is proceeding on a simple method for estimating the average density of the sediment collected in the centrifuge tube.

Columns 5 to 8 give a number of the parameters of the size distributions. Columns 9 and 10 were calculated using equations A-6 or A-8. Based on measurements on similar materials the porosity,  $\theta$ , was assumed equal to 0.3 in calculating column 10.

Comparison of the cohs per 1000 ft calculated from the size distributions, column 10, and the values measured photometrically, column 12, shows good agreement for all except the run on February 22. The mean coh value for column 10 is 1.28 and that for column 12 is 1.44, a difference of approximately 10 percent. The following discussion of a few of the uncertainties entering into the calculations shows that the agreement between the values in columns 10 and 12 is probably well within the experimental error.

While considerable information is available on light scattering coefficients for particles suspended in a transparent medium, none is available for particles in or on a filter as used in the conventional dust spot measurements. Therefore, certain simple experiments were carried out to estimate the accuracy with which the light scattering theory for transparent media could be applied to particles on a diffusing filter medium.

One simple experiment is illustrated in Table 5. Optical densities were measured with the dirty surface toward and away from the light source. A study of column 6 shows that the millipore filters which have all the particles lying essentially in one plane at the surface may exhibit considerably different light transmission values

TABLE 4—SUMMARY OF SIZE ANALYSIS DATA ON ATMOSPHERIC DUST SAMPLES TAKEN IN THE PARTICLE TECHNOLOGY LABORATORY

DATE	TIME P.M.	LENGTH OF RUN HOURS	(1) METHOD OF COLLECTION	(2) ASSUMED GM/CC	(3) COLOR OF SEDIMENT	(4) CONCENTRATION MG/M <sup>3</sup>	(5) $d_g$ MICRONS	(6) $\sigma_g$	(7) $d_n$ MICRONS
2/20	1:00	6	Note 1	2.1	Lt Brn	0.096	4.3	2.47	0.65
2/22	6:00	6	Note 1	2.1	Dk Brn	0.172	7.9	1.66	0.95
2/23	5:30	6	Note 1	2.1		0.061	2.3	3.36	0.49
2/25	9:45	5	Note 1	1.6		0.098	4.6	3.11	0.66
2/25	5:00	8	Note 2	1.6		0.102	3.3	2.56	0.54
2/27	6:30	6	Note 3	1.6	Black	0.081	2.4	3.16	0.62
2/28	4:30	4½	Note 3	1.6	Black	0.121	3.1	2.75	0.88
2/28	10:00	6	Note 3	1.6	Black	0.080	2.1	2.60	0.59
3/1	5:30	6	Note 3	1.6	Note 4	0.049	1.8	3.27	0.46

## MICROSCOPE DATA

	6 min.	Note 3				0.77	1.91	0.32
	6 hrs.	Note 3				0.78	1.85	0.30

## SEDIMENTATION DATA

DATE	(8) $d_n$ MICRONS	(9) NOTE 5	(10) COHS PER 1000 FT	(11)	(12)
				AISI SAMPLER	
				MEDIA	COHS PER 1000 FT
2/20	2.60	10000	1.24	Paper 4	0.46
2/22	5.10	4400	0.48	Paper 4	3.27
2/23	1.54	16300	0.83	Paper 4	0.75
2/25	2.58	10800	1.16	Paper 4	1.14
2/25	2.01	13800	1.58	Paper 4	1.48
2/27	1.61	19100	1.72	Paper 4	2.05
2/28	2.35	11200	1.48	Paper 4	1.44
2/28	1.68	17000	1.54	Paper 4	1.31
3/1	1.13	27000	1.50	Glass Fiber	1.10

## MICROSCOPE DATA

2/8	0.56		1.83		
2/23	0.62		0.98	Paper 4	0.75

Note 1.—Wet electrostatic collection; Note 2—Dry electrostatic collection; Note 3—Millipore collection; Note 4—Indicates sediment ½ brown and ½ black; Note 5— $\Sigma(K \varphi_1) (1/d_1)$  in square centimeters per cubic centimeter.

under different conditions. The data shown in this table and other similar studies with the microscope using objectives of varying numerical aperture have led to the following conclusions:

1. The ratio of the optical density with the dirty surface away from the light to that with it toward the light decreases with increasing penetration of particles into the filter,

TABLE 5—LIGHT TRANSMISSION STUDIES ON DIFFERENT FILTER MEDIA WITH TWO DIFFERENT PHOTOCELLS

(1) FILTER MEDIA	(2) MEASURED AFTER RUNNING-MIN.	(3) PHOTOCELL <sup>a</sup> ARRANGE- MENT	(4) OPTICAL DENSITY DIRT AWAY	(5) OPTICAL DENSITY DIRT TOWARD	(6) OD AWAY
					OD TOWARD
HA Millipore.....	15	A	0.040	0.024	1.67
HA Millipore.....	30	A	0.066	0.046	1.44
HA Millipore.....	60	A	0.119	0.093	1.28
HA Millipore.....	180	A	0.301	0.254	1.19
HA Millipore.....	180	B	0.308	0.193	1.60
Paper 41.....	15	A	0.024	0.024	1.00
Paper 41.....	30	A	0.046	0.047	0.98
Paper 41.....	90	A	0.104	0.104	1.00
Paper 41.....	90	B	0.103	0.099	1.04
Glass Fiber.....	15	A	0.193	0.169	1.14
Glass Fiber.....	30	A	0.327	0.298	1.10
Glass Fiber.....	15	A	0.195	0.172	1.13
Glass Fiber.....	30	A	0.351	0.321	1.09

<sup>a</sup> A = Photometer head as shown in Fig. 3. B = 60 mm diameter photocell lying in contact with the filter media.

decreases with increasing optical density and decreases with decreasing effective numerical aperture of the photocell.

2. The optical density of the filter with the dirty surface toward the light is probably the closest to and within a factor of about 1.4 of that which would be predicted from light scattering theory for transparent media.

TABLE 6—CALCULATION OF EFFECTIVE SCATTERING AREA FOR ATMOSPHERIC DUST ANALYZED BY SEDIMENTATION. SAMPLE COLLECTED 2/23/55

(1) d MICRONS	(2) % UNDERSIZE BY VOLUME	(3) MIDPOINT OF SIZE RANGE MICRONS	(4) % IN RANGE	(5) 1/d CM <sup>-1</sup>	(6) AREA PER SIZE RANGE (4) X (5)	(7) K (M = 1.5)	(8) SCATTER- ING AREA-CM <sup>2</sup> (6) X (7)	(9) % SCATTER- ING AREA LESS THAN SIZE
0.25	0.1	0.225	0.1	44500	45	0.90	40	0.25
0.30	1.0	0.275	0.9	36300	327	1.04	340	2.3
0.35	3.0	0.325	2.0	30800	616	1.67	1030	8.7
0.40	5.0	0.375	2.0	26650	531	2.25	1190	15.9
0.50	9.2	0.45	4.2	22200	918	2.96	2720	32.6
0.70	13.0	0.60	3.8	16780	642	4.15	2670	49.0
1.0	16.9	0.85	3.9	11750	458	3.83	1760	59.8
1.5	31.0	1.25	14.1	8000	1130	2.50	2820	77.1
2	40.7	1.75	9.7	5720	554	2.18	1210	84.6
4	67.7	3.0	27.0	3330	900	2.00	1800	96.6
8	75.4	6.0	7.7	1660	128	2.00	256	97.1
14	93.8	11.0	18.4	910	167	2.00	335	99.3
20	98.5	17.0	4.7	588	59	2.00	118	99.9
25	99.2	22.5	0.7	444	4	2.00	8	99.96
30	100.0	27.5	0.8	358	3	2.00	6	100.0
							16303 <sup>a</sup>	

<sup>a</sup> Figure is  $2K \int_0^1 1/d_1 = 16303 \text{ cm}^2 \text{ per cm}^3$ .

The second conclusion is also supported by another simple experiment that was performed to see if the assumed index of refraction of 1.5 was reasonably close to the true value. In order to obtain the  $K$  values tabulated in column 7 of Table 6 from Fig. 7, it is necessary to assume a value of  $m$ . From microscopic observation

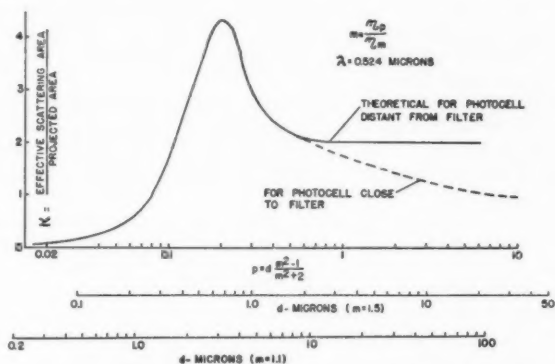


FIG. 7. UNIVERSAL LIGHT SCATTERING CURVE (AFTER SINCLAIR<sup>8</sup>)

$\eta_p$ , and hence  $m$  in air, was estimated as 1.5. If this is near the correct value, measurements of the optical density of the particles on a millipore filter immersed in a liquid, having a refractive index near that of the particles, should be less by a calculable amount than for the same filter in air, providing also that the measured size distribution is correct. In Table 7 are some measurements made in air, and in 4 different liquids together with the calculated ratios for the microscope and

TABLE 7—OPTICAL DENSITY OF ATMOSPHERIC DUST ON A MILLIPORE FILTER IMMERSED IN DIFFERENT FLUIDS

LIGHT MEASURING INSTRUMENT	FLUID	$\eta_m$	OD AWAY	OD TOWARD
As shown in Fig. 3 . . . . .	Air	1.000	0.077	0.074
Microscope and Photometer Inst. . . .	Immersion Oil	1.515	0.0313	
Microscope and Photometer Inst. . . .	Benzene	1.501	0.0334	
Microscope and Photometer Inst. . . .	Carbon Tet.	1.463	0.0310	
Microscope and Photometer Inst. . . .	Water	1.330	0.0386	
METHOD OF DETERMINATION		OD IN AIR OD IN LIQUID		
Measured (average for 4 liquids)			2.21	
Calculated from Microscope ( $m = 1.1$ )			2.16	
Calculated from Sedimentation ( $m = 1.1$ )			2.07	

sedimentation size distributions as given in Tables 1 and 6. The relatively good agreement indicates that the assumed refractive index of 1.5 is approximately correct, that the variation in  $K$  with  $p$  given in Fig. 7 is reasonably correct and that the measured size distributions are reasonably close to the effective size distribution of the particles on the dry filter and hence to the distribution of air-borne aggregates. This latter conclusion was further supported by microscopic observation of the particles resuspended after the sedimentation size distribution had been determined. The size and shape of the aggregates appeared to be the same as those on a dry millipore filter.

It will be noted in Fig. 7 that the highest  $K$  value occurs at a particle size of about  $3 \mu$  for  $m = 1.1$ . This  $m$  value corresponds approximately to that of the dust suspended in acetone. This is confirmed by the fact that the suspension in the capillary of the sedimentation tube appears the most dense at about 1 to  $3 \mu$  during the analysis.

From these data, certain important conclusions can be drawn regarding the meaning and usefulness of each of the 3 methods of measurement used in this study.

Fig. 4 and Table 1 illustrate some of the limitations of the microscope method of measurement of the size distribution of air-borne dust. In Fig. 4 it will be noted that 72 percent by number of the particles counted lie at or below the limit of resolution, which means that the number distribution is probably considerably in error.

The accuracy of a weight or volume distribution calculated from the microscope number distribution is even more open to question because of the large volume of a few large particles relative to the volume of the small particles. However, due to the sharp decrease in  $K$  for both large and small particles, it is possible to calculate the equivalent optical density of dust spots with reasonable success from the microscope size distribution.

The limitations of the sedimentation method are not quite as numerous as for the microscopic methods, but must be kept in mind in interpreting size distribution measurements. Perhaps the largest uncertainty arises from the necessity of assuming a mean density in calculating Stokes' equivalent spherical diameter.

From Stokes' Law, it is possible to derive the following expression for the error in calculated particle size for given errors in particle density.

$$\% \text{ error in } d = 100 \left[ 1 - \sqrt{\frac{\rho_{\text{true}} - \rho_0}{\rho_{\text{assumed}} - \rho_0}} \right]$$

For example, if  $\rho_{\text{true}} = 1.6$ ,  $\rho_{\text{assumed}} = 2.1$ ,  $\rho_0 = 0.8$ , the percent error in  $d$  would be 22 percent.

Since the projected area of the particles is proportional to the square of the particle diameter, the error in the cohs per 1000 ft calculated from the size distribution would be:

$$\text{Percent error in cohs per 1000 ft} = 100 [1 - (\rho_{\text{true}} - \rho_0)/(\rho_{\text{assumed}} - \rho_0)]$$

For the values assumed, the percent error in the calculated cohs per 1000 ft would therefore be 38 percent. In practice, it is believed that the density of the dust can probably be estimated to within 15 percent. Under this condition, the error in particle size would be about 10 percent, and the error in calculated cohs per 1000 ft, about 20 percent.

A second limitation of the sedimentation method arises from errors in the cumulative size distribution curve of the fine particle size caused by compaction of the sediment column and the smallest increment in sediment height which is readable. For example, in Fig. 4 the volume distribution curve by sedimentation is dashed below  $0.3 \mu$  because the minimum readable increment corresponds to about 1 percent of the total sediment height.

In view of the known uncertainties in the various measurements and calculations, it may be concluded that the agreement between the calculated and observed cohs per 1000 ft in columns 10 and 12 Table 4 is within the experimental error. If this is the case, it is possible to draw a number of interesting conclusions regarding the meaning of the dust spot method when used to measure atmospheric dust concentration.

Fig. 8 is a plot of the effective scattering area contributed by each particle size range of the volume distribution curve, by sedimentation shown in Fig. 4, relative

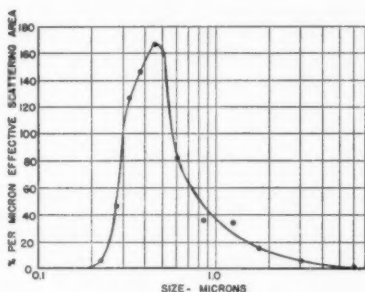


FIG. 8. RELATIVE EFFECTIVE SCATTERING AREA OF THE VARIOUS SIZE RANGES OF THE SAMPLE COLLECTED FEBRUARY 23. ANALYSIS BY SEDIMENTATION

to the total scattered by the dust. From this curve and from columns 2 and 9 of Table 6, it is apparent that the optical density of the dust spot is due primarily to particles between about  $0.3$  and  $1.5 \mu$  size. From Table 6 it may be noted that the 40 percent by volume of particles between  $0.3$  and  $2 \mu$  contributes 83 percent of the total optical density. It may also be noted from Fig. 7 that the rapid decrease in  $K$  for small particle sizes means that the dust spot method is not sensitive to particles below about  $0.3 \mu$ , if white light is used.

This may be a possible explanation for the relatively good agreement between the calculated and observed coh values. The microscopic method is insensitive to these small particles, because they are below the limit of resolution; the sedimentation method is inaccurate, because there is only a small percent by volume below this size, and the photometric method is insensitive to these small particles, because of the rapid decrease in  $K$ .

Thus, there is a need for a practical method of size distribution measurement below the  $0.3 \mu$  size range. There is some hope that the sedimentation method could be useful in this range if proper techniques were developed by comparison with the electron microscope.

From Fig. 7 it is seen that for an  $m$  value of 1.5, the peak in  $K$  occurs at about  $0.7 \mu$ . This means that the dust spot method is most sensitive to particles of this size. However, the peak in the effective scattering area of the dust sample shown in Fig. 8 occurs at about  $0.5 \mu$  because of the relatively large volume fraction of dust particles between about  $0.3$  and  $1 \mu$ .

Though the nine analyses shown in Table 6 certainly do not represent the range of size distribution and concentration of atmospheric dust that may exist, these data do show that considerable variation in size distribution may occur. Visual observation of the color of the sediment in the sedimentation has also revealed that the character of the dust is variable. For example, the sample taken on February 20 was collected during a very high south wind. The sediment for this analysis was a very light brown in contrast with the black color exhibited on days when the wind velocity was low and was blowing from the industrial area of Minneapolis, Minn.

Since the procedure here described is intended for the analysis of the finer particle sizes, a separate method for the evaluation of lint and other large particles is under development. When the lint evaluation procedure has also been developed, it is planned to use the combined techniques to make a study of the size distribution and concentration of air-borne dust under a variety of atmospheric, industrial and residential conditions.

#### ACKNOWLEDGMENTS

The authors wish to acknowledge the assistance of W. E. Engebretson, graduate student in mechanical engineering, in carrying out much of the experimental work reported in this paper, and to George F. Landgraf, Trion, Inc., for assistance in the design of and for the construction of the electrostatic precipitator used in this study.

#### APPENDIX

*Derivation of relationship between number and volume particle size distribution and the optical density of a dust spot:* In order to compare particle size distribution and concentration measurements with discoloration, it is necessary to calculate the equivalent discoloration from the size distributions. A convenient unit for comparison is the coh per 1000 ft of air suggested by Hemeon<sup>5</sup>.

$$\text{cohs per 1000 ft} = \frac{10^5 \times \log_{10} \frac{I_0}{I}}{\frac{qt}{A}} = \frac{10^5 \times \log_{10} \frac{I_0}{I}}{l} \quad \dots \quad (\text{A-1})$$

Thus, the cohs per 1000 ft are actually the hundredths of a unit of optical density accumulated on the filter per 1000 linear ft of air drawn through the filter.

For spherical particles, the optical density for particles of diameter  $d$  is given by the following<sup>7</sup>:

$$\log_{10} I_0/I = 0.433 K N_v d^3 l \quad \dots \quad (\text{A-2})$$

Substituting Equation A-2 into A-1 yields

$$\text{cohs per 1000 ft} = 4.33 \times 10^4 K N_v d^3 \quad \dots \quad (\text{A-3})$$

The total scattering coefficient,  $K$ , is a complex function of the ratio of the particle diameter to the wave length of the light used, the absorption, the refractive index, the

shape of the particle, the geometry of the optical system used, as well as other factors. However, Sinclair<sup>8</sup> has shown that for a given wave length, to a good approximation,  $K$  can be expressed as a function of  $d [m^2 - 1/m^2 + 2]$  where  $m$  is the relative refractive index of particle to suspending medium. This function is plotted in Fig. 7. Since  $K$  depends on particle size, it must be included in the formulas for calculating the total effective scattering from the size distributions.

# NOMENCLATURE

$A$ = area of filter.	$n_i$ = number of particles in class interval $i$ .
$A_o$ = area of filter counted by microscope.	$OD$ = optical density = $\log_{10} I_o/I$
$d$ = particle diameter — microns, centimeters or feet, as noted.	$q$ = sampling rate, volume per unit time.
$d_g$ = geometric mean of number size distribution.	$Q$ = total volume of air sampled.
$d'_g$ = geometric mean of volume or weight size distribution.	$t$ = elapsed time.
$d_n$ = arithmetic mean of number size distribution.	$V_p$ = total volume of particles collected = $(1 - \theta) \times$ sediment volume in centrifuge tube.
$d_v$ = arithmetic mean of volume size distribution.	$V_o$ = apparent sediment volume in centrifuge tube.
$d_s$ = size of particle of average projected area.	$v_i$ = volume fraction of particles in class interval $i$ .
$I$ = intensity of light transmitted through a dirty filter.	$\rho$ = particle density, grams per cubic centimeter.
$I_o$ = intensity of light transmitted through a clean filter.	$\eta_p$ = refractive index of particle.
$K$ = total scattering coefficient.	$\eta_m$ = refractive index of surrounding medium.
$K_1$ = scattering coefficient for particles of size $i$ .	$\theta$ = sediment porosity = 0.26 for spheres.
$l$ = linear feet of air sampled = $Q/A$ .	$\sigma_g$ = geometric standard deviation of a particle size distribution =
$m = \eta_p/\eta_m$	
$N$ = total number of particles counted during a microscope count.	
$N_v$ = number of particles per unit volume of air.	

$$\sqrt{\frac{d \text{ at } 84\%}{d \text{ at } 16\%}}$$

**Volume Distribution:** Since the effective scattering area per particle is  $Kd^2$  and the volume per particle is  $\pi/6 d^3$ , the effective scattering area per unit volume of particles of size  $d$  is  $(6 K/\pi) (1/d)$ , or for a size distribution is

$$(6/\pi) \sum K_1 v_i 1/d_i$$

therefore

$$N_v K d^2 = (V_p/Q) (6/\pi) \sum K_1 v_i (1/d_i) \quad \text{. . . . . (A-4)}$$

and Equation A-3 becomes

$$\text{cohs per 1000 ft} = 4.33 \times 10^4 (V_p/Q) (6/\pi) \sum K_1 v_i (1/d_i) \quad \text{. . . (A-5)}$$

Or since  $V_p = V_o (1 - \theta)$ , Equation A-5 finally becomes

$$\text{cohs per 1000 ft} = 8.28 \times 10^4 [V_o(1 - \theta)/Q] \sum K_1 v_i (1/d_i) \quad \text{. . . (A-6)}$$

Equation A-6 has been derived for spherical particles. Although no  $K$  values are available at the present time for other particle shapes, Ellison<sup>9</sup> indicates that most irregular particles can probably be treated as spheres of equivalent volume.

Table 6 illustrates how the  $\sum K_1 v_i (1/d_i)$  is calculated from the volume-size distribution determined by sedimentation.

**Number Distribution:** For a number size distribution determined by microscope count the average effective scattering area per particle is given by  $(\Sigma K_i n_i d_i^2)/N$  and, therefore,

$$\text{cohs per 1000 ft} = 4.33 \times 10^4 N_v (\Sigma K_i n_i d_i^2)/N \quad \dots \quad (\text{A-7})$$

Since  $N_v = (N/Q) (A/A_0)$  Equation A-7 becomes

$$\text{cohs per 1000 ft} = 4.33 \times 10^4 (A/A_0 Q) \Sigma K_i n_i d_i^2 \quad \dots \quad (\text{A-8})$$

Table 1 illustrates the method for calculating  $\Sigma K_i n_i d_i^2$  from the microscope number distribution.

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3. Air Sampling with Membrane Filters, by M. A. First and L. Silverman (*AMA Archives of Industrial Hygiene and Occupational Medicine*, Vol. 7, 1953, p. 1).
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## DISCUSSION

C. N. DAVIES,† London, England, (WRITTEN): The sedimentation method of size analysis adopted is open to the objection referred to at the bottom of page 464, namely, that it disintegrates aggregates which exist in the airborne dust and measures their constituent particles. Presumably the microscope counts on the filters related to aggregates, though this important point is not mentioned. Another objection to the method is the assumption that the porosity of the sediment in the centrifugal tube is independent of particle size; it seems improbable that the coarse and fine fractions of dust, which are known to have shape factors varying over the size range, will pack in similar ways. It is also a little difficult to understand what goes on in the converging section of the tube just above the measuring section.

If it were desired to relate the microscopic counts to the sedimentation method, either considerably greater numbers ought to have been taken or the observation restricted to the particles below a certain size. It is clear that a ridiculous situation arises when a single 20  $\mu$  particle can influence the distribution so profoundly. A selective sampling technique would have avoided this difficulty.

† London School of Hygiene & Tropical Medicine.

In view of the foregoing remarks, agreement with the light scattering results would not be expected to be very good. The light scattering calculations are also open to considerable criticism, so that the figures in columns 10 and 12 of Table 4 are a good deal closer than would have been anticipated.

The remarks on page 471 about the effect of reversing the paper should be compared with Davies and Aylward†. The effect is due to diffusion of the incident light by the filter. When the particles are towards the light the rays which they scatter in a forward direction are all picked up and re-diffused by the filter. Screening is effective only within the angle of the incident beam.

If the particles are on the photocell side they screen light diffused from the filter over the entire angle of acceptance of the photocell which is greater than the angle of the incident beam; hence the screening is greater.

It is clear from Fig. 3, that the photocell accepts light scattered by the particles over a wide angle.

Fig. 7 is drawn incorrectly; when  $p$  is large  $K$  tends to unity for the photocell close to the filter and does not decrease to a smaller value; it is probable, also, that the refractive index correlation proposed by Van der Hulst is superior to that used, which is valid only for very small sized particles in the region of Rayleigh scattering.

The decrease of the scattering coefficient with increasing angle of acceptance is best calculated by diffraction theory for particles above  $1$  or  $2\ \mu$  and appears to be underestimated in Fig. 7. (See Fig. A). For particles with a high absorption coefficient it seems that  $K$  has been over-estimated in the neighborhood of the maximum.

The reference to Ellison in the Appendix, Reference 9, goes some way beyond what he actually wrote. It is true in the Rayleigh-Gans region, but the work in the report extends far beyond this and is mainly in the Mecke region where screening is on an area basis.

R. S. DILL, Washington, D. C.: I find, in looking over what I have done in the past, that I was chairman of the TAC on Air Cleaning for 7 years, so perhaps I should be expected to have some opinions but everybody wants an executive position, and for the last several years so far as air cleaners are concerned, that is what mine has been.

Now, my attitude toward Dr. Whitby at present is like that of President Lincoln toward General Grant. The president was asked why he liked the general as the leader of the armies, and he said, "Well, he fights." So I like what Dr. Whitby is doing because he attacks his subject from many different angles. But I don't know enough about his program and findings to comment on the individual points. I respect the judgment of Dr. Whitby and the Technical Advisory Committee, and I believe they are headed in a direction that may lead to valuable results. Also, I think Dr. Whitby knows the challenge here, which is eventually to develop information on which to base evaluations of air cleaning devices, in terms that are understandable and usable by engineers in every-day practice.

The investigation is in a field of major difficulty, in which new methods and new ideas may be of great potential value, and the information developed will be welcomed by all. But the ultimate practical problem of producing a test method for evaluating air cleaners realistically, and obtaining general acceptance and adoption of the method, will be an at least equally difficult undertaking, judging by past experience. Very sound and adequate proof will be needed to advance a proposed method against the resistance natural in a field already dominated by strong opinions, and in which different groups have made, and are now making, very considerable investments to accumulate useful test data, which unfortunately are not all obtained by the same method.

It is because of this embedded diversity that adoption of one generally acceptable method is important, and further, why it will require foresight, prudence, and a solid foundation of facts in the present undertaking to achieve success in adoption of a test

† The Photoelectric Measurement of Coal Dust Stains on Filter Paper, by C. N. Davies and Mary Aylward (*British Journal of Applied Physics*, Vol. 2, December 1951, pp. 352-359).

method. It is most encouraging that the Technical Advisory Committee has been able to support, and have the benefit of, the work that is being done by Dr. Whitby.

GEORGE LANDGRAF, Pittsburgh, Penna.: I have only one small comment. I would like to emphasize the correlation between the sensitivity of the light scattering method and the general mass media of dust which is found in the atmosphere. I believe Dr. Whitby's values have generally been confirmed by other workers. In other words, the sensitivity of the light scattering method seems to be highest between about seven-tenths and one and one-half microns, where one would expect to find the majority of atmospheric dust. I think that is an important point in the design of testing equipment.

S. F. DUNCAN, Los Angeles, Calif.: In listening to Dr. Whitby's talk, and particularly in connection with the colored slides he showed, it seems to me that the colored slides should be included in the paper, if at all possible, because it gives a great many of us that have not seen these little sedimentation tubes operate a chance to see what the deposit looks like.

Another point that has impressed me about the paper is that Dr. Whitby has taken more or less as his objective the determination of what exists as a dust problem in some possible location. This, of course, has been a problem of the committee under which he works. First, what is the dust problem, and then, possibly second, how can it be solved?

The one question that I have is that with respect to the computation of particle size from light scattered data, collected particles on either a standard filter paper or millipore filter paper, may overlap each other and look like an agglomerate to the photocell where they were not in the original air stream, and I have wondered if there is any way this effect can be either confirmed or denied.

I think Dr. Whitby is to be complimented on the wide front on which he has attacked the problem and that he has made a very sizable contribution to the literature.

J. W. MAY, Louisville, Ky.: I really do not feel sufficiently qualified to comment on this subject, but find it most interesting. One point that disturbs me to some extent is that the constituents of atmospheric dust, as well as the particle size, seem to vary with the season of the year, the direction of the wind, the geographical location of the point in question, and on other factors. In this paper the author states that he has been able to check the particle size determinations within 10 percent by using alternate methods, and I cannot help but wonder what practical use the data will have in attempting to secure that accuracy—when 15 minutes later we may have an entirely different set of conditions. I would like to have Dr. Whitby comment on this situation. To me it is a wonderful idea to be able to secure this basic type of data, but I would like to know what can be expected in the way of practical application.

DEAN L. E. SEELEY, Durham, New Hampshire: My question deals with the particles that may be soluble in water. What happened in a case like that, in connection with sedimentation? Those of you who were at the Surf Club yesterday must realize there were many tiny particles of salt in the air, resulting from the evaporation of the sea water, and I just wondered what happened.

AUTHORS' CLOSURE (K. T. Whitby): All I can say in response to Dr. Dill's kind comments is that I hope history justifies his confidence in our work.

I should like to again emphasize the point made by Mr. Landgraf in his last sentence. It is very important to consider what is actually being measured when the dust spot or discoloration arrestance of an air cleaner is being determined.

Unfortunately there is no practical way to reproduce the colored photographs of the sediment referred to by Mr. Duncan. The pictures which show the colors of different sediment layers have been very useful to us in making estimates of the nature and amounts of the different dust constituents. Further data of this type will be presented when the results of the current dust survey are available.

The Lambert-Beer Law relating concentration and light extinction used to derive Equation A-2 accounts satisfactorily for the effects of overlap in the optical density range used here.

Mr. May's reaction to the variability of airborne dust is natural and reasonable, but I cannot agree with him, that the situation is hopeless. I have confidence that when sufficient data are available on all of the various characteristics of airborne dust, many conclusions useful for engineering purposes will be forthcoming. At the time of writing of this closure (October 1955), useful information is already evolving from the dust survey data. Space and scientific caution does not permit a discussion of these conclusions at this time.

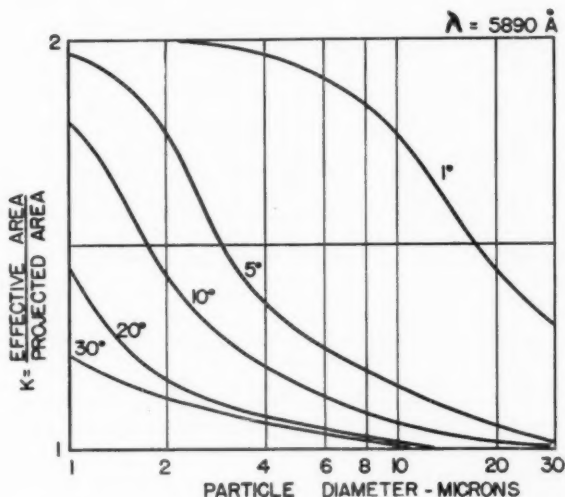


FIG. A—EFFECTIVE SCATTERING COEFFICIENT AS A FUNCTION OF SEMI ANGLE OF PHOTOCCELL ACCEPTANCE AND PARTICLE SIZE

Dean Seeley's question about the effect of particle solubility on the results is well justified. Actually acetone is being used for the sedimentation liquid. Acetone extraction on a large number of samples by Chambers, et al\* showed that the acetone soluble fraction averaged about 5 percent and was rarely over 15 percent by weight. We feel that this small error is far outweighed by other advantages of our procedure.

In addition to the discussions submitted to the ASHAE Meeting, another fine discussion was received by the authors at the request of Mr. Hemeon, from Dr. C. N. Davies. Permission was received by the authors to present this discussion here.

Dr. Davies brings up a number of technical points which are worthy of discussion, but which do not affect the validity of the conclusions drawn in the paper. It should be emphasized again that the agreement with 10 percent or so of the averages of columns 10 and 12, Table 4, does not imply that our conclusions are based on such agreement. As is pointed out by Dr. Davies, the magnitude of the various assumptions made in

\* The Characteristics and Distributions of Organic Substances in the Air of Some American Cities, by L. A. Chambers, E. C. Tabor and M. J. Foter (R. A. Taft Sanitary Engineering Center, U. S. Public Health Service, Cincinnati, Ohio).

the calculations is such that the best agreement that can be expected is within a factor of about 2. Why this level of agreement still constitutes a good overall check on the validity of the size analysis method can best be seen from Table 6. Note in this table that 49 percent of the total light scattering power of the dust is caused by only 13 percent by weight of the dust which is below  $0.7 \mu$ . Doubling the weight of dust below  $0.7 \mu$  would cause nearly 50 percent change in the light scattering power of the dust. The large effect on the optical density caused by a relatively small change in the size distribution, is the factor which makes this method of checking the size analysis useful.

The questions raised by Dr. Davies in the first paragraph of his discussion are discussed in Part II of the paper<sup>□</sup> on the centrifuge method.

Dr. Davies' suggestion that it would be better to use  $K$  values calculated from diffraction theory, rather than from light scattering theory is a good one. Subsequent work in our laboratory has shown that photometric methods essentially measure only the opaque particles on the filter. Therefore it is better to use the diffraction curves for opaque particles as shown in Fig. A for the calculations. However, this will reduce the magnitude of the calculated  $\text{coh}$  values in Column 10, Table 4, by only about 25 percent for most samples.

The effect of angular aperture as shown in Fig. A, must be accepted with considerable reservation. The curves shown in the figures are for particles in a transparent medium. Although studies are not complete of the effect of angular aperture, when the particles are suspended on a translucent filter material, the indications are that the  $K$  values do not decrease as rapidly as indicated in the figure, as the aperture is increased. While these effects do not affect the validity of the general conclusions drawn in the paper it appears that they must be taken into account whenever discoloration measurements are used for filter evaluation or other critical applications. The optical design of the photometer can have considerable effect on the relationship between the optical density and the dust load on the filter. At the present time a newly constructed high sensitivity null balance photometer is being used to study some of these factors. This work will be reported in a future paper.

Dr. Silverman of Harvard University has brought to the attention of the authors, the fact that Burke<sup>a</sup>, had devised a sedimentation method for the analysis of dusts deposited on millipore filters, and this work is credited at this time.

<sup>□</sup> This paper begins on page 449 of this volume.

<sup>a</sup> AITTA Quarterly, by W. C. Burke, Jr. (Vol. 14, 1953, p. 299).



**1550**

## SOURCES OF VENT GAS IN A HOT WATER HEATING SYSTEM†

By L. N. MONTGOMERY\*, BOSTON, MASS., AND W. S. HARRIS\*\*, URBANA, ILL.

IT IS a recognized fact that gases continue to accumulate in hot water heating systems even though the systems have been in operation for years. While industry has developed automatic air vent valves and other devices intended to reduce the necessity of periodic manual venting of the gas, there has been little study of the possible sources of this gas.

It was observed during the course of the regular research work at the I=B=R Research Home that it was necessary to vent the room heating units on a regular schedule to prevent air binding. Furthermore, it was noted that the most gas collected in the room heating units on the second story which were connected to the main near the boiler supply connection, and lesser amounts collected in the other second story units and the first story units near the boiler. Only traces of gas were ever vented from the other radiators in the system. This observation and previous work by others suggested that the mechanism of vent gas accumulation and its release was suited to experimental analysis.

### DESCRIPTION OF EQUIPMENT

The Research Home, shown in Fig. 1, was a two-story dwelling designed and furnished to represent a well-built American home. It was equipped with instruments to measure operating conditions of the heating system and temperatures produced and for recording room air and boiler water temperatures. Thermocouples

† This paper is a digest of a thesis submitted by the senior author in partial requirement for the degree of Master of Science in Mechanical Engineering in the Graduate College of the University of Illinois.

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

were provided for measurement of air temperatures throughout the house, and for measurement of water temperature at various points in the heating system. An elbow meter<sup>1</sup> was provided for measuring water flow rates in the system.

The same heating system was in continuous use during the time covered by this report. It was a one-pipe, forced-circulation hot water system, designed in accordance with I=B=R Installation Guide No. 100. The supply and return trunks were 1-in. iron pipe, and  $\frac{3}{4}$ -in. pipe was used for the mains in the north and south loops. All branch piping was  $\frac{1}{2}$ -in. A diagram of the piping layout is shown in Fig. 2. A design water temperature of 215 F was used to determine the amount of radiation installed in each room.



FIG. 1. I=B=R RESEARCH HOME

The only changes made to the heating system as a part of this investigative program were made during the summer of 1952. At that time, an 18 gal air cushion tank was substituted for the 24 gal tank in use during the 1951-52 heating season. At the same time, the point at which the air cushion tank was connected to the heating system was changed from the angle flow control valve above the boiler to a special dip-tube boiler fitting.

The 3-section, cast-iron boiler was fired by a single-port gas conversion burner using natural gas with a heating value of 1000 Btu per cu ft. The firing rate was maintained at 100 cu ft per hr, and the carbon dioxide content of the flue gas was held at 8 percent. Under these conditions the measured gross boiler output was 66,000 Btu per hr.

A diagram of the control circuit is shown in Fig. 3. The high limit control in the boiler stopped the burner at boiler water temperature of 235 F, but allowed the circulator to continue operating. The low limit control in the boiler was set to maintain the boiler water temperature above 165 F when the circulator was not operating. It was disconnected during some special tests.

<sup>1</sup> The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in a Pipe (University of Illinois, Engineering Experiment Station Bulletin 289, 1936).

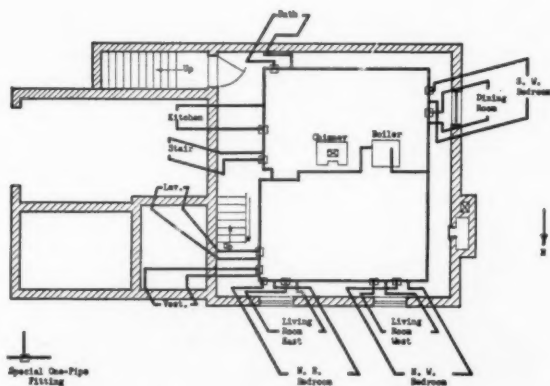


FIG. 2. DIAGRAM OF HEATING SYSTEM PIPING

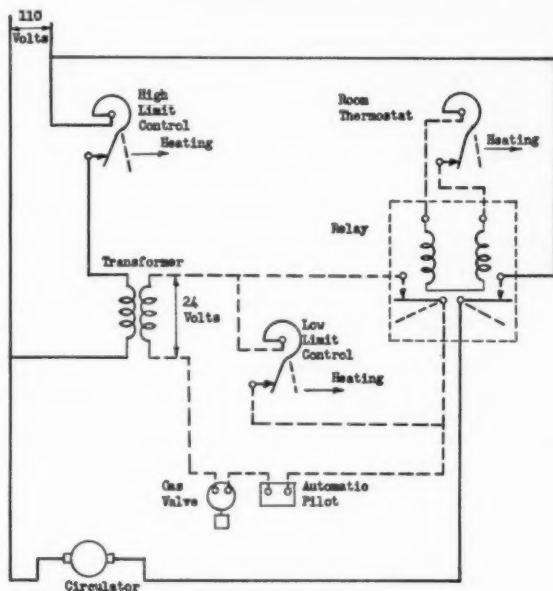


FIG. 3. DIAGRAM OF CONTROL SYSTEM

## ANALYSIS OF VENT GASES

A preliminary study of possible vent gas sources indicated that gases might be transferred from the air cushion tank by solution and later released from the water in the system; that they might be evolved from chemical reactions occurring in the system; and that they might enter the system as dissolved gas in the fill water.

A representative sample of gas vented from the system approximately 40 days after filling the system with tap water had the analysis shown in Table 1.

It was considered that the source of the carbon dioxide might be either dissolved gas in the fill water or dissolved bicarbonates which were being broken down into carbonates, gas and water. Some methane was known to be dissolved in the tap



FIG. 4. RUBBER BULB IN AIR CUSHION TANK

water used to fill the system, and it was assumed that this source accounted for all of the methane present. The hydrogen was probably released in a gas-evolving corrosion reaction which would dissolve iron in the process. The quantity of nitro-

TABLE 1—COMPOSITION OF GASES VENTED FROM CONVENTIONAL HOT WATER SYSTEM

EXTRACTED GAS		COMPOSITION PERCENT BY VOL.
Carbon Dioxide.....	CO <sub>2</sub>	1.1
Hydrogen.....	H <sub>2</sub>	12.8
Methane.....	CH <sub>4</sub>	2.2
Nitrogen.....	N <sub>2</sub>	83.0
Oxygen.....	O <sub>2</sub>	0.9
		100.0

gen, however, was too great to assume that it was dissolved in the water used for filling the system. It was evident that there must be a continuing source of this gas.

## AIR CUSHION TANK AS A VENT GAS SOURCE

The air trapped in the air cushion tank was a likely source of the nitrogen and oxygen observed in the vent gases. This air is intended to provide a relief for thermal expansion as the water in the main part of the system is alternately heated and cooled. Thus, water from the system enters the tank compressing the air slightly during the time the burner is in operation. The water in the air cushion tank was essentially at room temperature, and remained so because the mass of hot water transferred by expansion was small compared to the volume of the water already in the tank. Since gases are more readily soluble in cold water than in hot,



FIG. 5. AIR CUSHION TANK

some of the air in the tank was taken into solution by the water. As the system cooled some water containing dissolved air left and went back to the system. This process gradually increased the concentration of dissolved gas in the entire heating system until eventually the saturation point was reached at some point during periods of each heating cycle. Gases were then released in the system piping, the boiler or room units and carried through the system as bubbles to collect at high points.

An air tight rubber bulb, Fig. 4, was inserted in the tank through the fitting shown in Fig. 5. After insertion, the bulb was inflated to slightly above atmospheric pressure and the remaining space inside the tank was completely filled with water by venting all free air as the tank was filled. An air cushion was thus provided, but with no direct contact of air and water.

After the system had operated for two weeks the rate of gas release was measured and a sample taken for analysis. The rate of gas release when operating with no direct contact of air and water in the air cushion tank was reduced to roughly one-quarter of the rate obtained when operating with free contact between the water and the air. Table 2 shows the vent gas analysis for 2 successive bi-weekly tests.

The actual rate of release of hydrogen and methane in these tests had not changed to any marked degree from the rate of release before the rubber bulb was inserted in the air cushion tank. The reason for the percentage increase of hydrogen and methane reported in Table 2 as compared with Table 1 was that the total rate of release of nitrogen and oxygen had decreased sharply. The small amounts of nitrogen and oxygen which were observed were probably already in solution in the water of the system at the time the rubber bulb was installed in the air cushion tank. These tests verified the belief that nitrogen and oxygen were transmitted from the air cushion tank when there was an interface between the air and water in the tank.

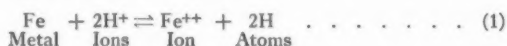
## CHEMICAL REACTIONS AS A SOURCE OF VENT GASES

*Corrosion Processes:* Both the hydrogen and the carbon dioxide which were reported in the gas analyses in Tables 1 and 2 were considered to be evolved by chemical reactions. The source of the hydrogen was assumed to be a corrosion

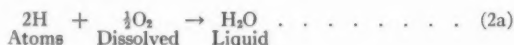
TABLE 2—COMPOSITION OF VENT GAS WHEN OPERATING WITH RUBBER BULB IN AIR CUSHION TANK

EXTRACTED GAS	COMPOSITION PERCENT	
	FIRST TEST	SECOND TEST
Carbon Dioxide.....CO <sub>2</sub>	1.3	0.5
Hydrogen.....H <sub>2</sub>	71.9	71.9
Methane.....CH <sub>4</sub>	2.0	9.8
Nitrogen.....N <sub>2</sub>	23.3	16.0
Oxygen.....O <sub>2</sub>	1.5	1.8
	100.0	100.0

reaction which dissolved iron in the water as ferrous hydroxide and released hydrogen at the iron surface. According to Speller<sup>2</sup> when an iron surface is exposed to water, the iron atoms and hydrogen ions formed in dissociation of water molecules will react to form ferrous ions and hydrogen atoms.



The ferrous ions will go into solution to replace the hydrogen ions and the hydrogen atoms will *plate out* or form a thin film on the exposed surface. Before the reaction can proceed any further, the hydrogen atoms must be removed. This can be accomplished by destruction of the film.



or its removal as bubbles of gas



For the case being considered, part of the ferrous ions from Reaction 1 remained in solution as ferrous hydroxide



and the remainder was deposited as a soft, black magnetic coating over the water-side of the system.

<sup>2</sup> *Corrosion, Causes and Prevention*, by F. N. Speller, (McGraw-Hill Book Co., Inc., New York, N. Y., 1951, p. 13-16.)

Whether Reaction 2a or 2b was predominant depended upon the acidity of the water and the amount of free oxygen present. From earlier tests it was determined that gas was taken into solution by the water in the air cushion tank and then distributed to all parts of the heating system. A material balance would be required to determine how much oxygen was so dissolved and what portion of this may have been used by Reaction 2a.

During the 1952-53 heating season, the heating system was filled with fresh water at the start of the winter, and the same water was used throughout the rest of the season.

The pH of the water used to fill the system initially was 7.60. Within two weeks, however, it rose to 8.0 and remained between 8.0 and 8.1 for the rest of the season. These pH values indicate the water in the system was at all times neutral or slightly alkaline, a condition not favorable for the occurrence of Reaction 2b.

TABLE 3—MATERIAL BALANCE SUMMARY

COLUMN	(1)	AIR CUSHION TANK				DIFFERENCE IN VOLUME, CU IN., COL. 3—COL. 5	VENTED GAS	
		START		END			COMPOSITION, PERCENT BY VOLUME	VOLUME, CU IN.
		COMPOSITION, PERCENT BY VOLUME	VOLUME, CU IN.	COMPOSITION, PERCENT BY VOLUME	VOLUME, CU IN.			
Carbon Dioxide.....	CO <sub>2</sub>	21.0	312	0.5	5	—5	0.5	1.1
Oxygen.....	O <sub>2</sub>	79.0	1172	6.9	72	240	1.9	4.2
Nitrogen.....	N <sub>2</sub>			92.6	960	212	97.3	212.0
Hydrogen.....	H <sub>2</sub>						0.3	0.7
		x =	1484	y =	1037	447		218.0

The material balance, also made during the winter of 1952-53, was started about one week after the system had been refilled following alterations. The quantity of air lost from the air cushion during normal operation of the heating system was determined by the change in water level of a gage glass on the air cushion tank and was compared with the quantity of gas vented from the system.

Over a period of 29 days, a total of 218 cu in. of gas was vented from the system. In the same interval the water level in the tank rose  $2\frac{1}{2}$  in., indicating the loss of about 447 cu in. of free air from the tank. The difference between the gas vented and the gas lost from the tank was (447 — 218) or 229 cu in. All pipe joints had been sealed and checked for leaks and precautions were taken to insure as nearly as possible the same amount of gas in solution in the system water at the beginning and end of the test. This indicated a loss of gas in some manner other than venting.

No sample of the air in the air cushion tank was taken at the start of this test, but there is no reason to doubt that it consisted of oxygen and nitrogen in about the same ratio as found in the atmosphere. This being true, the percentage composi-

tion of the gas in the air cushion tank at the start of the test was as shown in Table 3, Column 2. The percentage composition at the end of the 29-day test was determined by chemical analysis and is shown in Column 4. The percentage composition and the quantities of gases vented from the system during the 29-day test are shown in Columns 7 and 8, respectively. If it is assumed the water in the systems was saturated with nitrogen at the start and end of the test, then the quantity of nitrogen in the vent gases must equal the quantity of nitrogen absorbed by the water in the air cushion tank, or

$$0.79x - 0.926y = 212 \text{ cu. in.} \quad (4)$$

where

$x$  = free volume of gas in air cushion tank at start of test in cubic inches.

$y$  = free volume of gas in air cushion tank at end of test in cubic inches.

Also, the change in water level in the air cushion tank indicated that

$$x - y = 447 \text{ cu. in.} \quad (5)$$

Simultaneous solution of these equations yields a value of 1484 cu in. for  $x$  and 1037 cu in. for  $y$ . Having determined the values of  $x$  and  $y$ , the remaining values in Columns 3 and 5 of Table 3 were obtained using the corresponding percentage values in Columns 2 and 4. The oxygen which apparently combined with hydrogen was  $(240 - 4) = 236$  cu in. This figure checked closely with the difference in the volumes of gas lost from the air cushion tank and that vented from the system  $(447 - 218 = 229 \text{ cu in.})$ .

Knowing the quantities of oxygen which combined with hydrogen and free hydrogen remaining, Reactions 1, 2a, and 2b may be used to estimate the rate of corrosion in the system. Reaction 2a states that 1 volume of oxygen will combine with 2 volumes of hydrogen gas. Two hundred thirty six cu in. of oxygen were consumed by this reaction requiring 472 cu in., or 0.02269 oz of hydrogen. Reaction 1 states that the ionization of one atom of iron to the ferrous state will release 2 atoms of hydrogen. The atomic weights of iron and hydrogen are 55.8 and 1.0, respectively. Therefore, the 0.02269 oz of hydrogen entering into Reaction 2a represented the ionization of  $0.02269 \times 55.8/2 = 0.6330$  oz of iron. The 0.7 cu in. (0.00003147 oz) of hydrogen formed by Reaction 2b represents the ionization of 0.0009 oz of iron. Over the 29-day test period a total of  $0.6330 + 0.0009 = 0.6339$  oz of iron were dissolved. Thus it appears that the rate of iron corrosion in the system was only a few ounces per year.

Water velocities in the system were fast enough to insure a uniform loss of metal with no pitting of the pipe walls; therefore, the rate of corrosion indicated by the evolution of small amounts of hydrogen should be no cause for alarm. It should be noted that the release of hydrogen gas in very small amounts will not cease, but will be a continuing process in any system. The very small extent to which even neutral water is ionized is sufficient to allow Reaction 2b to proceed at a slow rate.

While the generation of hydrogen by Reaction 2b was of interest, it was not and is not normally the method by which really destructive corrosion occurs. Reaction 2a is generally the most active and is dependent upon dissolved oxygen for completion. It has been found that the rate of metal corrosion in neutral or slightly alkaline solutions is directly proportional to the concentration of dissolved oxygen. Therefore, corrosion will not be a problem in hot water heating systems so long as

the concentration of dissolved oxygen in the water is maintained at a low level, and the water is maintained neutral to slightly alkaline.

*Breakdown of Dissolved Solids as a Source of Vent Gas:* When the heating system was being filled at the start of the 1952-53 heating season, a sample of the fill water was taken for partial analysis with the following results:

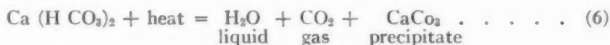
TEST	REPORTED AS	PPM
Iron (total).....	Fe	0
Alkalinity (P).....	CaCO <sub>3</sub>	0
Alkalinity (M).....	CaCO <sub>3</sub>	304
pH.....		7.60

Within 2 weeks after the system was put into operation the pH of the system rose to 8.0 and remained between 8.0 and 8.1 for the rest of the second season.

At the end of the season, another partial analysis was made with the following results:

TEST	REPORTED AS	PPM
Iron.....	Fe	0.9
Alkalinity (P).....	CaCO <sub>3</sub>	2
Alkalinity (M).....	CaCO <sub>3</sub>	64

The water supply for the twin cities of Champaign-Urbana comes from local wells, and it contains a large amount of calcium bicarbonate in solution compared to the calcium carbonate present. This is indicated by the relation between the phenolphthalein (P) alkalinity test, which gives a measure of the carbonates present and the methyl orange (M) test which gives a measure of the bicarbonates. All of the alkalinity in the fill water was due to bicarbonates which impart a *temporary* hardness to the water. Upon heating, these bicarbonates decompose to release water, carbon dioxide, and calcium carbonate, some of which is precipitated. The equation for this process is:



Equation 6 was the source of carbon dioxide observed in vent gases.

Not all of the calcium carbonate which was released in the breakdown of bicarbonates in the system was precipitated. Some of the carbonate remained in solution, as evidenced by the P alkalinity at the end of the season. Where P is less than  $\frac{1}{2}$  M, carbonates in solution are calculated to be 2 P, or, in this case 4 ppm. Bicarbonates accounted for the remainder of the M alkalinity; hence, bicarbonates present were  $(64 - 4) = 60$  ppm. The value of 60 ppm was less than one-fifth the amount of dissolved bicarbonates present at the beginning of the season.

Carbonates in solution hydrolyze to form acid ions to a lesser degree than bicarbonates, and if other conditions of the water which affect pH remained the same, breakdown of the bicarbonates to carbonates offers a satisfactory explanation for the observed increase in pH.<sup>3</sup>

The amount of iron present in the water at the end of the heating season was not enough to affect the turbidity of the water, and was probably present as ferrous hydroxide.

#### FILL WATER AS A DIRECT SOURCE OF VENT GAS

The water used to fill the heating system was a direct source of vent gas, since measurable quantities of oxygen, nitrogen, and methane were dissolved in the tap water. The total water content of the system used during the 1952-53 heating season was 28 gal, including the water in the air cushion tank. When this water entered the system, it contained approximately 0.8 parts per thousand (ppt), or 5 cu in., of methane, 12.5 ppt, or 81 cu in., of nitrogen, and 25 ppt, or 162 cu in., of oxygen. Since there was no source of methane other than the fill water, it served as a tracer, and its disappearance indicated that the gas dissolved in fill water had been completely vented.

Analyses of the vent gas were made at intervals during the heating season, and traces of methane were still observed 4 months after beginning operation. However, the analysis after 6 months of operation showed no trace of methane, indicating that the fill water was only a temporary source of vent gas.

#### CONCLUSIONS

1. The main source of vent gas in a hot water heating system is nitrogen which is transferred from the air cushion tank. Metal corrosion which evolves hydrogen gas in small quantities is a secondary and continuing source. Temporary sources are gas dissolved in the make-up water or evolved from the breakdown of dissolved solids.

2. Oxygen in the make-up water and in the air cushion tank will combine chemically in a corrosion reaction, resulting in a drop in the static head of a closed system. This corrosion reaction is minor from the standpoint of metal consumed.

#### ACKNOWLEDGMENT

Experimental work to determine the sources of vent gas in a hot water heating system was performed at the I=B=R Research Home during the winters of 1951-52 and 1952-53. The authors wish to acknowledge the assistance of Dr. T. E. Larson and R. M. King, head of the Chemistry Subdivision and assistant chemist, respectively, of the Illinois State Water Survey. These men acted in an advisory capacity and assisted in gas and water analyses. Acknowledgment is also made to those manufacturers who supplied the equipment used.

<sup>3</sup> *Corrosion Causes and Prevention*, by F. N. Speller (McGraw-Hill Book Co., Inc., New York, N. Y., p. 384).

## DISCUSSION

F. N. SPELLER, Pittsburgh, Penna. (WRITTEN): I have read the paper on "Sources of Vent Gas in a Hot-water Heating System", and note that this interesting paper is based on tests in a *closed* system with an air cushion tank just above the boiler. However, it may be advisable to note by way of discussion that in the common *open* system with a vented expansion tank at the top, the gas accumulated in radiators is mostly hydrogen.

In a tight system the dissolved oxygen is soon fixed and as very little, if any, make up water is required, there is only about 3 percent of the corrosion found on steel pipe in a hot water *supply* system, and that is due to the evolution of hydrogen gas. In my residence in Pittsburgh the accumulation of this gas in an open hot water system, has been practically stopped by the addition of a little alkalinity to the water so that it has a pH about 9. This treatment requires very little attention.

H. A. LOCKHART, Morton Grove, Ill.: I would like to ask Professor Harris if there has been any consideration given to evaluating conditions of vent gases when the pH in the system was on the acid side. There has been a certain amount of information which, I believe, would indicate that the type of gases and the frequency of those gases would be entirely different if one did not have an alkaline water, that is if it were on the acid side.

JOHN W. JAMES, Chicago, Ill.: I have no particular questions to direct to Professor Harris, but thought the western members of the Society would like to know how the subject matter of this paper fits into the general Society Research Program.

The research work of the Society is supported by the membership to the extent of 40 percent of our dues, and this is supplemented by substantial industry contributions. A continuation of the work which Professor Harris explained this morning is now in progress at Urbana in the form of a typical hot water heating system installed in a laboratory. When the Research Technical Advisory Committee of the Society, concerned with hot water and steam heating, learned of the work which was described by Professor Harris, this morning, it decided that this subject was worthy of further investigation. Subsequently, the Committee took the initiative to request the Committee on Research to appropriate funds which you, in part, contribute to the Society each year, to develop a laboratory study at Urbana. In general, this system consists of a typical boiler installation with a one-pipe forced hot-water system, with radiation above and below the main to simulate typical conditions in the field. The purpose of this study is not only to study sources of vent gases in a system of this kind, but also to investigate where the gases accumulate and to endeavor to measure the amount of accumulated gases with some thought as to how they may be effectively eliminated.

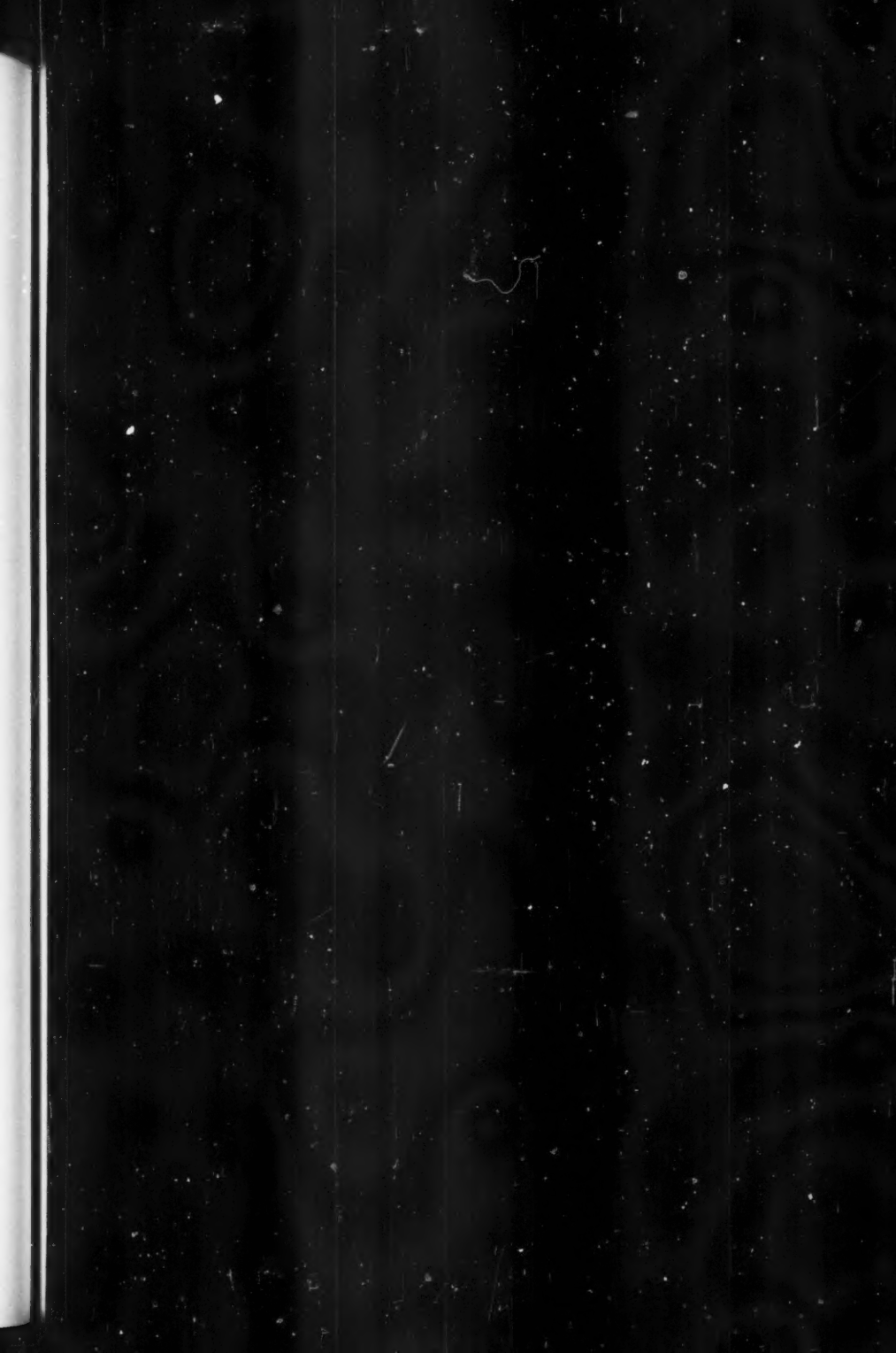
At a recent meeting of the TAC on Hot Water and Steam Heating in Evanston, Professor Harris submitted some data regarding this investigation, and whereas cast iron radiation has been used in the first part of this study, conventional convectors are being installed to study further the problem of vent gas accumulation.

I thought this information would be of interest to give a practical example of the manner in which your research money is being expended.

AUTHORS' CLOSURE (Professor Harris): As Mr. Speller has commented, the vent gases that appear in an open system, in which the expansion tank must be located above the highest radiator, will be primarily hydrogen. It might be added that this would also be true in a closed system if the compression tank were located at the high point so that it would have low pressure as well as low temperature.

Mr. Lockhart and Mr. Speller touched upon the importance of the pH value as to the quantity and composition of the gases that will be released from the system. The formation of hydrogen ions by electrolytic corrosion of metal, and then the conversion of hydrogen ions directly to gas, rather than combining with oxygen, is a characteristic reaction in acid water. Therefore, one would anticipate a greater corrosive action and a greater amount of hydrogen from this particular source if the pH value was lower than

those we were using; however, we have made no tests to prove this. It should be pointed out that the pH value alone is not a complete index of the corrosiveness of the water. A pH value of 8 to 10 represents the minimum corrosiveness, and if there is no oxygen present it represents virtually no corrosion. However, if dissolved oxygen is present in the water corrosion may result due to the fact that the hydrogen ions can combine with this oxygen to form water. There would be no gas evolved in that reaction, but nevertheless corrosion would occur.



**1551**

## PSYCHROMETRIC ANALYSIS FOR DESIGN OF FORCED DRAFT AIR COOLING TOWERS

By S. E. AGNON\*, HAIFA, ISRAEL, AND B. H. SPURLOCK, JR.\*\*<sup>1</sup>, BOULDER, COLO.

**T**HE COOLING process in a tower may be expressed with the aid of the psychrometric chart (Fig. 1). The chart is a combination of a humidity chart and an enthalpy diagram, both having a common abscissa divided into Fahrenheit degrees.

An important characteristic of the enthalpy part of the chart is that the scale of the ordinate divided into Btu is equal to the scale of the abscissa divided into Fahrenheit degrees. Since the abscissa may be considered to represent the heat content of the water (with specific heat of water equal to unity) and also since the abscissa has the same division as the ordinate, a direct geometrical and visual comparison between the enthalpies of air and water is possible. The *operating line* (to be taken up later) can be found by direct measurement, eliminating the necessity of tedious calculations.

The enthalpy may be closely represented<sup>1</sup> by

$$i_m = C_p T + 1061 H + 10 \dots \dots \dots (1)$$

where

$i_m$  = enthalpy of air-vapor mixture, Btu per pound dry air.

$T$  = air temperature, degrees Fahrenheit.

$H$  = specific humidity, pounds of water per pound dry air.

$C_p$  = specific heat of dry air at constant pressure, and at temperature  $T$ , Btu's per pound per Fahrenheit degree.

A simple way to find the enthalpy of the air mixture  $i_m$  is to locate the point on the psychrometric chart by means of the dry- and wet-bulb temperatures. Then follow the wet-bulb temperature line until the saturation line is reached, thence

\* Mechanical Engineer.

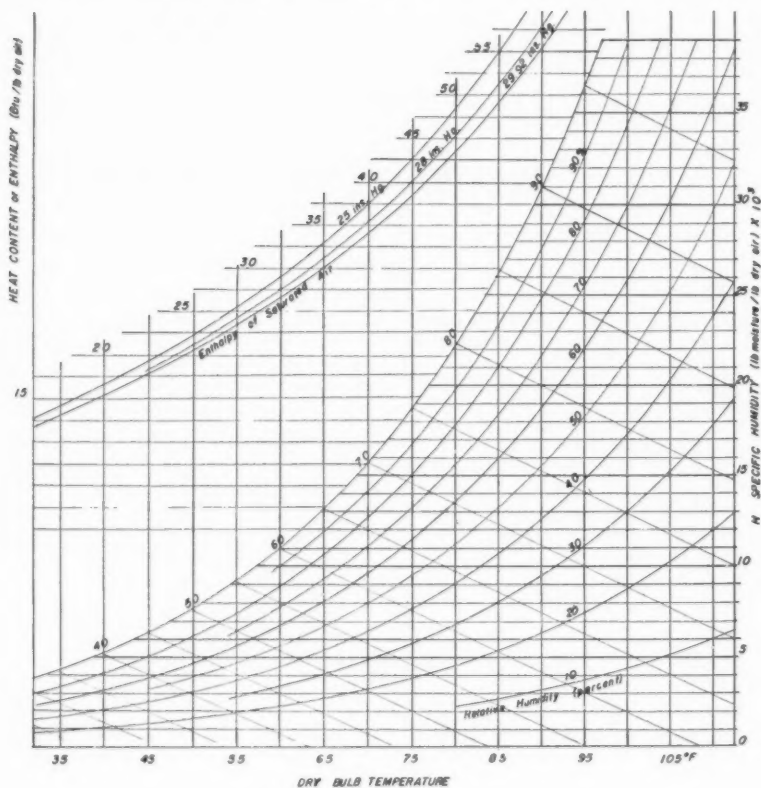
\*\* Professor of Mechanical Engineering, University of Colorado, Member of ASHAE.

<sup>1</sup> Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, Calif., June 1955.

vertically upward until the enthalpy curve is reached. The value is read on the ordinate (Fig. 2, line A-a'-a''-a). In order to satisfy different barometric pressures, 3 separate curves are included in Fig. 1.

Any cooling or heating process between a dry surface and the air is represented by a horizontal line (Fig. 2, line A-B), whereas a cooling process as a result of heat transfer between the air and a wetted surface will lie on a straight line connecting



line processes (A-B), (A-C), and (A-D) can be reproduced on the enthalpy diagram by (a-b), (a-c), and (a-d). For practical purposes these are straight lines.

The enthalpy difference between any two points (X) and (Y) of any process shown on the humidity chart (Fig. 2) can be expressed as:

$$\Delta i_m = i_{mx} - i_{my} = C_{pav} (T_x - T_y) + 1061 (H_x - H_y) \quad \dots (2)$$

where  $C_{pav}$  is the mean value of the specific heat between  $T_x$  and  $T_y$ . Dividing Equation 2 by  $(T_x - T_y)$ ,

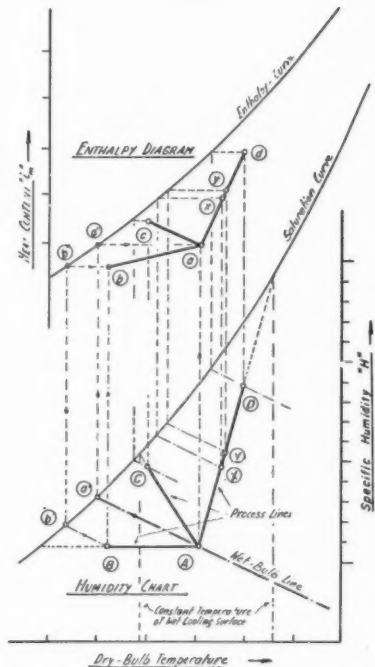


FIG. 2. CONSTANT TEMPERATURE COOLING PROCESS

$$(i_{mx} - i_{my}) / (T_x - T_y) = C_{pav} + 1061 (H_x - H_y) / (T_x - T_y) \quad \dots (3)$$

The term  $(H_x - H_y) / (T_x - T_y)$ , representing the slope of a straight line on the humidity chart drawn between the points (X) and (Y), is a fixed value and

$$(i_{mx} - i_{my}) / (T_x - T_y),$$

being the expression for the slope of the process lines on the enthalpy chart is also fixed and the enthalpy process line on the chart as well as the process line on the humidity chart is a straight line between any 2 points (X) and (Y).

## TOWER COOLING PROCESS

Such is not the case for a process when the temperature of the surface is varying. Such an example is the cooling tower process. The cooling tower process may be represented by some curve such as (A-B) in Fig. 3, or its corresponding enthalpy curve (a-b).

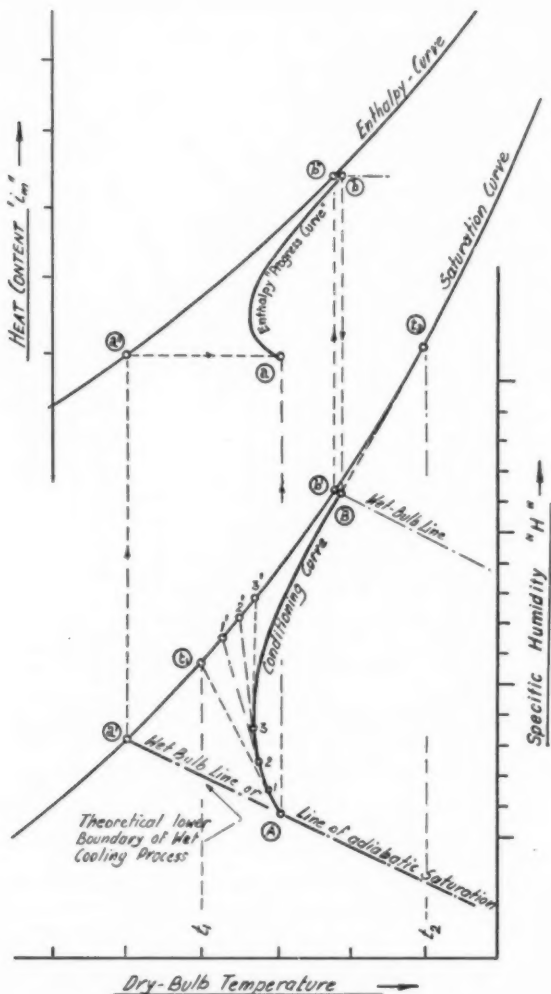


FIG. 3. TOWER COOLING PROCESS

In order to construct these curves it is necessary to review the fundamental principles involved in the cooling tower.

A cooling tower is a chimney-like device (Fig. 4), where air and water are brought into contact, the air moving up or across the tower while the water falls by gravity. In the cooling tower, the problem is to cool water while in an evaporative cooler the air is cooled. The fundamentals are the same. During the cooling process

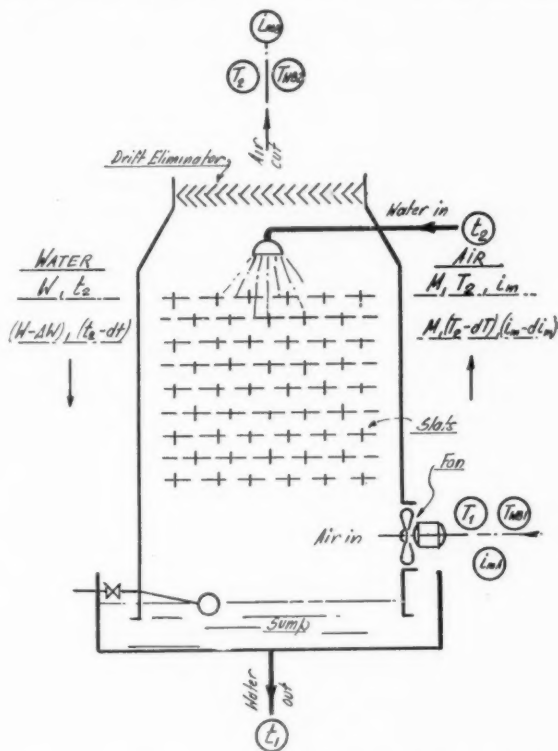


FIG. 4. SCHEMATIC ARRANGEMENT OF COOLING TOWER

not only is there a sensible heat exchange, but also a latent heat exchange in the water evaporated. It follows that the process must always proceed in the direction of an increase in humidity and will be accompanied by an increase in enthalpy of the air (not necessarily a rise in temperature). (If chilled water is sprayed into warm air the enthalpy of the air-vapor mixture might decrease.) The evaporative process establishes the temperature of adiabatic saturation (wet-bulb temperature lines, Fig. 3) as the lowest boundary of any such water cooling process and in general fixes the wet-bulb temperature as the theoretically lowest obtainable water temperature.

In contrast to the previously described process where the surface temperature is considered uniformly constant, both the air and the liquid surface temperatures vary.

At any instant, at any elevation in a vertical tower, the temperature may be assumed uniform and constant and the heat exchange, during this instant, will take place along a line connecting the supply condition (1) with a point (1') on the saturation curve at the corresponding water temperature (see Fig. 3).

A little higher in the tower, an instant earlier in respect to the water stream and an instant later in respect to the air flow, the water temperature will be higher by a small amount, whereas the air condition will have moved from (1) a small distance along the line (1-1') until it reaches point (2). The point fixing the condition of the air at (2) will move along the line (2-2') until point (3) is reached, and so on.

It is evident that the curve of the cooling tower process or *conditioning curve* is the *envelope* of individual *net process* lines as just discussed. For construction purposes, the conditioning curve may be regarded as a succession of small individual straight lines (A-1), (1-2), (2-3), (3-4), and so on. Each of these individual elements has its corresponding counterpart in the enthalpy diagram, all of which together form a curved line illustrating the air enthalpy throughout the tower (see Fig. 3). This curve, called the *progress cooling curve*, shows clearly the principle of increasing enthalpy during the water cooling process. The progress cooling curve may be used in the same manner as the conditioning curve for calculations without the necessity of constructing the conditioning curve. In the Appendix under *Mathematics of the Cooling Tower Process* it is shown that the conditioning curve as well as the humidity part of the chart is unnecessary and does not affect the accuracy of the method. However, as shown there, the slope of the operating line,

$$\tan \phi = \Delta i_m / \Delta i = W/M \dots \dots \dots (4)$$

cannot be selected arbitrarily, but must be determined by considering the problem of cooling tower performance.

#### COOLING TOWER PERFORMANCE

As mentioned, the lowest water temperature that is theoretically obtainable coincides with the wet-bulb temperature of the entering air. In practice, temperatures a few degrees above the wet-bulb temperature are common and the difference is termed the *approach* (see Fig. 5 and Fig. 6, which is a detail of Fig. 5). The smaller the approach the higher the thermal efficiency of the tower. The efficiency may be defined as

$$\eta_t = \text{Cooling range} / (\text{Cooling range} + \text{approach}) \dots \dots \dots (5)^\dagger$$

which, in Fig. 5 corresponds to

$$\begin{aligned} G-A' &= \text{Cooling range} \\ G-a'' &= \text{Cooling range} + \text{approach} \end{aligned}$$

The numerical value of the approach depends on the design of the tower. A value of 5 to 6 deg Fahrenheit is the practical minimum, and it can be postulated that

<sup>†</sup> For considerations concerning the meaning and use of Equation 5 see Appendix under *Comments Concerning Equation 5*.

$$t_1 \geq \text{wet-bulb temperature} + 5 \text{ F} \dots \dots \dots (6)$$

From Equation 4 it is evident that the slope (tan  $\phi$ ) of the operating line and the water-air ratio ( $W/M$ ) are dependent on each other. In order to keep the air requirements, the fan size and the power as small as possible, theoretically the

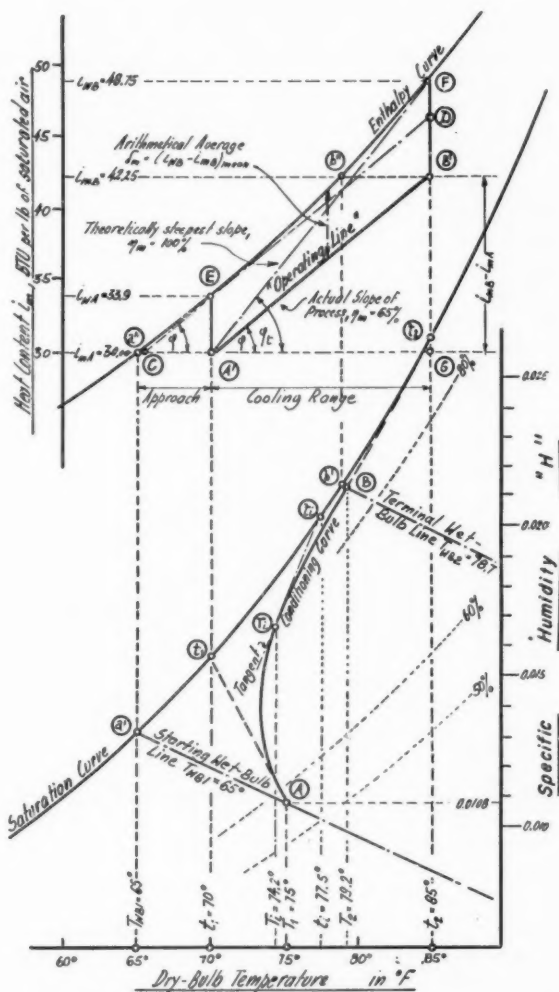


FIG. 5. GRAPHICAL ANALYSIS OF COOLING PROCESS

operating line should be selected so that it passes through point (A') and tangent to the enthalpy curve at (F) (Fig. 5). This would require an infinite tower height. A flatter line gives economical height but must be steep enough to keep the water-air ratio within practical limits and not reduce the mechanical efficiency of the tower too much. A distinction is made between the thermal efficiency  $\eta_t$  and

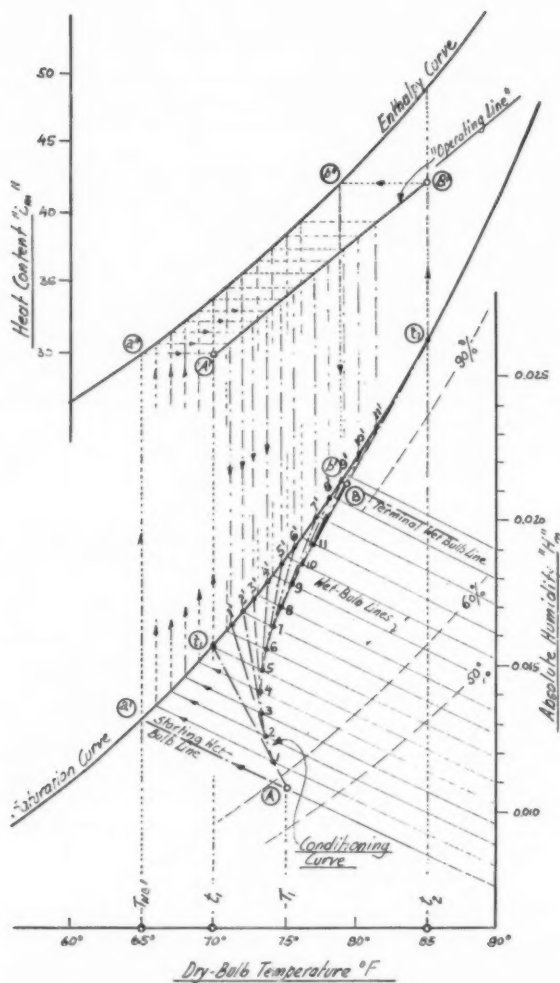


FIG. 6. CONSTRUCTION OF CONDITIONING CURVE

the mechanical efficiency  $\eta_m$ . The mechanical efficiency is the ratio of the theoretical minimum amount of air  $M_t$  to the actual amount of air  $M$  required for the cooling process at the same thermal efficiency  $\eta_t$ . That is

$$\eta_m = (M_t/M) = (M_t/W)/(M/W) = \tan \phi / \tan \phi_t \quad \dots (7)$$

where  $\tan \phi$  is the slope of the operating line for the actual cooling process and  $\tan \phi_t$  is the slope of the operating line for the infinite tower. (See Appendix under *Minimum Air at Infinite Tower Height*).

The mechanical efficiency  $\eta_m$  is influenced by the tower design, and in addition to some extent by the thermal efficiency  $\eta_t$ . When determining the angle  $\phi$ , Mc-

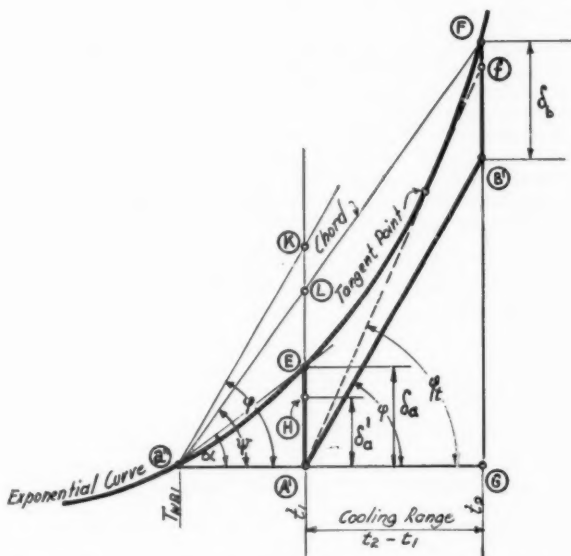


FIG. 7. GEOMETRICAL RELATIONS (CURVATURE OF ENTHALPY CURVE IS GREATLY EXAGGERATED)

Adams<sup>2</sup> points out that the mechanical efficiency  $\eta_m$  should not exceed 75 percent, as with already high values of  $\eta_t$  any small increase in  $\eta_m$  will require an excessively large increase in tower height, without a corresponding gain in  $\eta_t$  (see Fig. 9).

In most cases the mechanical efficiency

$$\eta_m = \tan \phi / \tan \phi_t$$

can be measured directly by the length ratio  $(B'-G)/(F-G)$  as is evident from Fig. 5. However, caution must be employed, not to do so indiscriminately as will be seen in the extreme case demonstrated by Fig. 7. In this case (Fig. 7) the mechanical efficiency must be expressed by means of Equation 7, and is  $(B'-G)/(f-G)$ .

The reason for the difference in the expression is due to the fact that the tangent point in the latter case (Fig. 7) is at a lower temperature than  $t_2$  and not coincident with or higher as for most cases.

The range of the water-air ratios is, therefore, dependent on the tower design and this must be taken into consideration when selecting the slope. In practice the  $\tan \phi$  can assume any value between 0.6 and 1.40. In the majority of cases, the value may be assumed between 0.9 and 1.1. For a rough approximation and an average value of  $\tan \phi$ , a value equal to unity may be assumed conforming to a slope angle of 45 deg.

Having fixed the terminal conditions of the cooling process and selected the slope of the operating line, it becomes possible to determine more-or-less accurately, the condition curve as shown in the following problem.

*Problem: A cooling tower is to be designed to cool 120 gpm of water over a temperature range of 15 F to the lowest obtainable temperature.*

*Outside air conditions are  $T_1$  equal to 75 F, and  $T_{wb1}$  equal to 65 F, corresponding to a relative humidity of approximately 57 percent. The mechanical efficiency of the tower is assumed to be 65 percent and the approach is 5 F.*

*Draw the conditioning curve, estimate the amount of air required for cooling and calculate the thermal efficiency. Find the temperature  $T_2$ , as well as the wet-bulb temperature  $T_{wb2}$  of the air at the top of the tower and determine the temperature  $T_1$  of the air at a point in the tower, where the water has been cooled by half of the temperature range.*

*Solution:* (See Figs. 5 and 6, keeping in mind that Fig. 6 is a detail of Fig. 5)

The lowest obtainable temperature of the water at the bottom of the tower is according to Equation 6

$$t_1 = T_{wb1} + 5 = 65 + 5 = 70 \text{ F}$$

Since the cooling range is 15 F, then the water must enter the top of the tower at 85 F. The thermal efficiency is according to Equation 5:

$$\eta_1 = \text{Cooling range} / (\text{Cooling range} + \text{approach}) = 15 / (15 + 5) = 0.75$$

To construct the conditioning curve it is necessary to draw the operating line. Fix the point (A) on the humidity chart (Fig. 5) from the given data. Follow the constant wet-bulb temperature line to the saturation line (point a'), then vertically upwards to the enthalpy line (point a'') and horizontally to the right until it intersects the temperature  $t_1$  (point A'). This is the point (A) on the humidity chart reproduced on the enthalpy chart. Now draw a straight line through (A') to the point (F) where the initial water temperature ( $t_2$ ) crosses the enthalpy curve. By measurement the slope of (F-A') is

$$\tan \phi_1 = 1.28.$$

Then from Equation 7, the slope of the operating line is

$$\tan \phi = \eta_m \tan \phi_1 = 0.65 \times 1.28 = 0.835$$

The operating line is constructed with a slope

$$\phi = \tan^{-1} 0.835 = 40 \text{ deg};$$

through point (A') until it intersects the temperature  $t_2$  at the point (B'). The operating line is now known and the conditioning curve may be constructed.

Draw a straight line from point (A) (refer to Fig. 6 for detail) to the point  $t_1$  on the saturation curve. Select a point (1) on the line (A- $t_1$ ), a small distance from (A). Proceed from (1) parallel to a constant wet-bulb temperature line to the saturation

curve, thence vertically to the enthalpy curve, then to the right to the operating line and vertically downwards to the saturation curve. This fixes the point (1'). Draw a line from (1) to (1'). On this line (1-1') close to (1) select a point (2) and repeat the same procedure as before locating the point (2'). This method is followed until the temperature  $t_2$ , the initial temperature of the entering water, is reached.

To find the terminal conditions of the air, simply reverse the method. Starting at (B') on the enthalpy chart, proceed horizontally to the left to (b''), thence downwards to (b') and along a constant wet-bulb temperature line to the point (B) on the conditioning curve. This gives the conditions for the outgoing air to be 79.2 F for dry-bulb temperature and 78.7 F for wet-bulb temperature (Fig. 5).

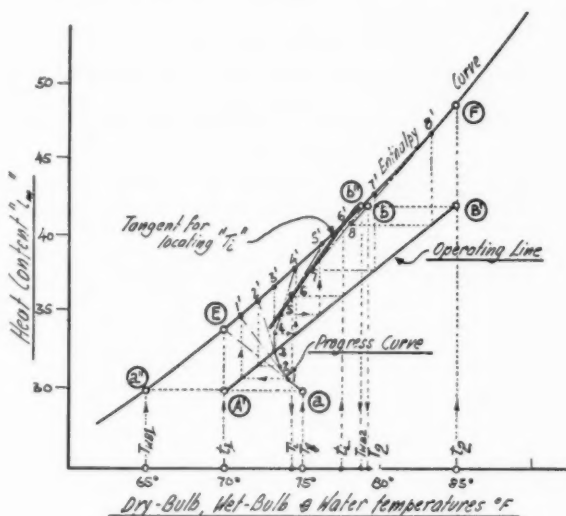


FIG. 8. CONSTRUCTION OF PROGRESS CURVE

The curve enveloping the individual points (A, 1, 2, 3, 4, etc.) represents the conditioning curve and illustrates the path of the cooling process. The smaller the elements, the more exact are the results.

In order to find any 2 corresponding temperatures of air and water during the process it is necessary only to draw a tangent to the conditioning curve. The point of tangency represents the air temperature  $T_1$ , the intercept at the saturation curve indicates the water temperature  $t_1$  (Fig. 5). The water temperature  $t_1$  at half the cooling range is

$$t_1 = (85 + 70)/2 = 77.5 \text{ F}$$

From the point  $t_1$  equal to 77.5 on the saturation line a line is drawn tangent to the conditioning curve. This is the temperature  $T_1$  equal to 74.2 F corresponding to the air conditions in the tower where the water temperature is 77.5 F.

The amount of water to be cooled is 120 gpm or 1,000 lb per min. Therefore the amount of air required for cooling is by Equation 4

$$M = W/\tan \phi = 1000/0.835 = 1200 \text{ lb per min, and} \\ (1200 \text{ lb/min})/(0.075 \text{ lb/ft}^3) = 16,000 \text{ ft}^3/\text{min standard air}$$

The same results could have been obtained by means of the progress curve. The construction of this curve is similar, but actually it is simpler than that of the conditioning curve.

Locate points (A') and (a) as indicated previously. Keeping in mind that every process line of the humidity chart has its counterpart on the enthalpy chart, draw a line (a-E) (See Fig. 8), corresponding to (A-t<sub>1</sub>) on Fig. 6. Now select a point (1) on the line (a-E) close to (a), proceed horizontally to the operating line, then upwards to the enthalpy curve at point (1'). Connect point (1) and (1'). Choose point (2) close to point (1) on the line (1-1') and go through the same process as before, locating point (2'). Continue until the progress curve is completed.

To find the intermediate temperature  $T_1$ , the method is exactly the same as with the conditioning curve.

### TOWER HEIGHT AND AREA

The principal value of the operating line lies in its importance for estimating the height of the cooling tower. It was shown in a previous paper<sup>3</sup> that the *overall driving force or potential* at any point in the tower may be represented by

$$\delta = (i_w - i_m),$$

which is the enthalpy of the saturated air minus the enthalpy of the air at the same temperature. This is the vertical distance between the operating line and the enthalpy curve.

If a line is drawn through point (A') and tangent to the enthalpy curve at point (F) Fig. 5, it is noted that the potential difference becomes smaller towards the top of the tower and in this case (infinite tower) would approach zero as a limit. As was indicated, smaller values of  $\phi$  must be selected for economical reasons.

The tower height can be expressed as a function of the area (A'-E-F-B') in Fig. 5. If  $Z_{og}$  is called the *height of a mass transfer unit*, then the total effective height of the cooling tower can be found by the equation

$$Z = Z_{og} \int_{i_{MA}}^{i_{MB}} \frac{di_m}{i_w - i_m} = N \cdot Z_{og} \quad \dots \quad (8)$$

where

$$N = \int_{i_{MA}}^{i_{MB}} \frac{di_m}{i_w - i_m},$$

and is called the number of *mass transfer units*.

The value of  $Z_{og}$  for use in Equation 8 is a function of  $(m/Ka)$ , where  $m$  is the mass velocity of the air measured in lb per min per sq ft and  $Ka$  expresses the *overall mass transfer coefficient* in the same units for a specific type of tower packing. The values of  $Ka$  are a function of both the water velocity and the air velocity. In some cases higher flow rates require higher towers, in others, the reverse. Experimental values of  $Z_{og}$  are available in the literature.

In Table 1 representative data are given for various types of tower packing. The data indicate the average value of  $Z_{og}$  over their range of validity.

With values of  $Z_{og}$  thus available, the remaining part of solving Equation 8 consists of finding ways of arriving at the value of  $N$ . There are several possible methods of procedure.

One of these is by a step-wise, or graphical, integration of Equation 8. Equations for such a solution are developed in the Appendix under the heading *Graphical Integration of Equation 8* and by means of an example this method is illustrated there.

Another method is also described and illustrated in the Appendix† which permits a somewhat more rapid procedure in making the calculations. This method is based on the equation:

$$N = \frac{\eta_m}{2} \left( \frac{1}{1 - \eta_t} + \frac{1}{1 - \eta_m} \right) + \frac{0.12 \eta_m \eta_t \tan \phi}{(1 - \eta_m)(1 - \eta_t)} \quad (9)$$

TABLE 1—DATA FOR FORCED DRAFT COOLING TOWERS

TYPE OF TOWER PACKING	$Z_{og}$ , ft	MASS VELOCITY OF AIR;	MASS VELOCITY OF WATER;
		$m = \text{LB/SQ FT MIN}$	$w = \text{LB/SQ FT MIN}$
Vertical Spray Tower <sup>a</sup>	10 — 2.5	7	3.3 — 10
Vertical Spray Tower <sup>a</sup>	4.7	5 — 12	7
Grid Packing <sup>a</sup>	14.4	9.2	12
Grid Packing <sup>a</sup>	19	17 — 38	53
Spindled tubing, horizontal staggered <sup>a</sup>	3.5 — 3.9	7.2 — 12.7	11.7
Raschig Rings, 15 mm <sup>b</sup>	1 — 0.9	7.5 — 15 <sup>c</sup>	d
Streamlined Wood Sections, 2.3/4 in. long <sup>b</sup>	5 — 7	9.5 — 17 <sup>c</sup>	d

<sup>a</sup> Reference 2, p. 290.

<sup>b</sup> HEATING, VENTILATING, AIR-CONDITIONING GUIDE 1947, p. 670.

<sup>c</sup> These mass velocities refer to the free (not the gross) area in the tower.

<sup>d</sup> Complete wetting of the packing surfaces is supposed.

#### THE NOMOGRAPH (FIG. 9) FOR DESIGN

None of the methods of evaluating  $N$  just mentioned is wholly simple and consequently it seems desirable to work out a graphical means of making this solution by the use of a straight edge and a minimum of computations.

To construct a chart for this purpose Equation 9 is plotted in the form of an alignment chart along with a second alignment chart expressing  $\tan \phi$  in terms of known basic data, viz:

$$\tan \phi = \frac{\eta_m}{\eta_t} \cdot e^{\left( \frac{T_{WB}}{40} + 1.75 \right)} \left[ e^{\left( \frac{t_2 - T_{WB}}{40} \right)} - 1 \right] \quad (10)^*$$

The result of this plotting is the chart of Fig. 9.

Two barometric pressures (29.92 and 25.0 in. Hg) have been provided on Fig. 9. One may interpolate between these 2 pressures with the only error involved at this point. Comparison of the 2 scales shows that a falling barometer has the same effect as increasing the wet-bulb temperature with constant barometer and vice versa. For example, the 65 F at 29.92 in. Hg corresponds to 61.3 F at a barometer of 25 in. Hg.

*Problem: As an example of the use of the chart, let us solve for  $N$  when  $\tan \phi = 0.835$  and  $W = 1000$  lb of air per min, and when the mechanical efficiency ( $\eta_m$ ) is 0.65 and the thermal efficiency ( $\eta_t$ ) is 0.75.*

† Under the heading *Development and Use of Equations 9*.

For the mathematical analysis leading to Equation 10 see the Appendix under *Development of Equation 10*.



## OPERATION UNDER CHANGING SETS OF CONDITIONS

Often a cooling tower will have to be operated under conditions different from those for which it was designed. In the absence of performance data the problem of estimating the comparable performance is not easy, even by means of the detailed graphical analysis. However, the nomograph Fig. 9 handles this problem readily.

To demonstrate the difficulties encountered, consider a tower operating with a varying wet-bulb temperature. It will be remembered that the cooling process depends on the position, length and slope of the operating line, while the height of the tower is determined by the *shape* of the area of the driving forces. Let the process (I) in Fig. 10 represent such a process and (II) represent the process after the

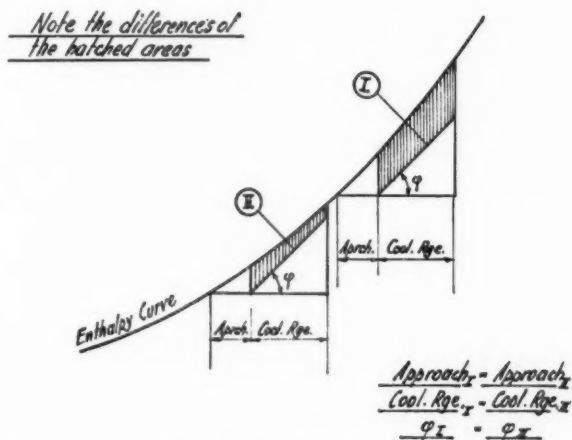


FIG. 10. OPERATION UNDER CHANGING CONDITIONS (CURVATURE OF ENTHALPY CURVE IS GREATLY EXAGGERATED)

wet-bulb temperature has been changed. It is seen that if the operating data maintained, a modification takes place in the shape of the area of the driving forces, involving a higher mechanical efficiency and a greater number  $N$ .

For an existing tower, this is obviously impossible, the number  $N$  being fixed and the same irrespective of temperature conditions<sup>†</sup>, consequently some of the operating data must change in order to give the same tower height. The nomograph eliminates extensive *trial and error* calculations and the time spent is appreciably reduced. The following typical problem illustrates the use of (Fig. 9)

<sup>†</sup> This statement is valid only in the case of constant air-water flow rate. For certain types of towers the height of the mass transfer unit  $Z_{OG}$  is a function of the flow rates (See Table 1), so that in such a case the number  $N$  cannot be assumed constant under all circumstances. However within the range of flow variations caused by weather variations, the number  $N$  may be assumed constant.

*Problem:* The following data are obtained from a cooling tower; wet-bulb temperature 55 F, rate of water flow 528 gpm, inlet temperature of water 95 F, outlet temperature of water 70 F, air flow rate 4,000 lb. per min (53,400 cfm standard air).

*If the wet-bulb temperature is changed to 60 F and the tower performance is to remain the same, what modification in the tower is necessary? (Note: Tower inlet temperature of the water  $t_2$  is to remain the same).*

*Solution:* The first step towards solving the problem consists of finding the number  $N$  of mass transfer units of the tower by means of Fig. 9. For this purpose begin by computing  $\eta_t$  and  $\tan \phi$ .

From Equations 5 and 4

$$\eta_t = (t_2 - t_1)/(t_2 - T_{WB}) = (95-70)/(95-55) = 25/40 = 0.6$$

and

$$\tan \phi = W/M = (528 \times 8.33)/4000 = 1.1$$

On Fig. 11, with a wet-bulb depression (cooling range plus approach) of 40 F and a water-to-air ratio of 1.1, draw the lines 1-2-3-5. The ratio

$$\eta_m/\eta_t = 1.10$$

is obtained, from which

$$\eta_m = 1.1 \times 0.6 = 0.66.$$

From

$$\eta_m = 0.66, \eta_t = 0.60$$

and point (5) by means of a straight edge point (7) is located and

$$N = 2.2.$$

This is the number of mass transfer units which remains constant for the tower.

The following steps are almost the reverse of the foregoing, but with a different wet-bulb temperature. Raising the wet-bulb temperature to 60 F and keeping the temperature of the inlet water  $t_2$  at 95 F will cause a reduction in the approach, thereby raising the thermal efficiency.

Through the wet-bulb temperature 60 F and the wet-bulb depression (cooling range plus approach) of 35 F extend the line to point (3') (Fig. 11). Connecting this point (3') with the previous water to air ratio

$$W/M = \tan \phi = 1.10, \text{ (lines } 1'-2'-3'-5), \text{ thereby giving}$$

$$\eta_m/\eta_t = 1.05, \text{ at the point } (4_1).$$

Using the line (5-7) locate the point (6<sub>1</sub>) on the line

$$\eta_m/\eta_t = 1.05.$$

This corresponds to values of

$$\eta_m = 0.65, \text{ and}$$

$$\eta_t = 0.62$$

The cooling range is equal to

$$\eta_t \times (t_2 - T_{wb}) = 0.62 (95 - 60) = 21.7 \text{ F}$$

as compared to 25 F, even though the thermal efficiency has increased from 60 to 62 percent.

In order to handle the same heat load as before, the amount of water circulated will have to be increased as well as the air rate to maintain the constant water-air ratio

assumed. The increase in rate of flow of the two fluids is  $(25-21.7)/21.7 = 0.15$ , and the pump and fan will have to be modified accordingly.

It is perhaps more desirable to maintain the water rate constant and change the air

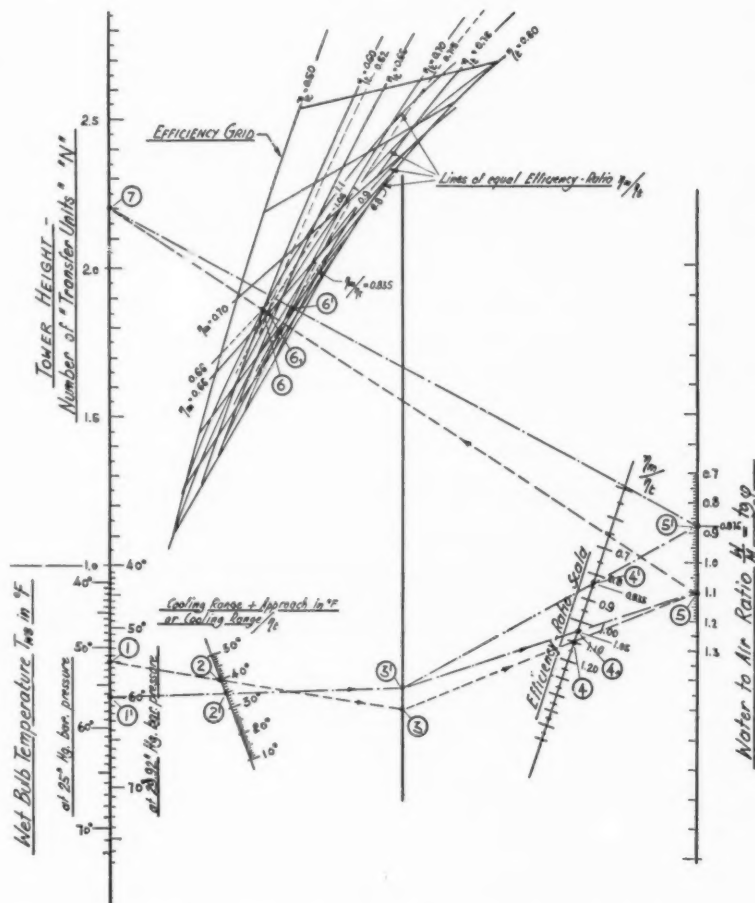


FIG. 11. SKELETON NOMOGRAPH SHOWING SOLUTION OF PROBLEM OF OPERATING CONDITIONS DIFFERENT FROM DESIGN CONDITIONS

rate to accommodate this. This step consists of keeping the cooling range at 25 F and modifying the water-air ratio ( $\tan \phi$ ).

In this case

$$\eta_t = 25/35 = 0.715$$

The point (5') indicating the new value of  $\tan \phi$  will be found by *trial* and must be so located that the line (3'-5') and the  $\eta_m/\eta_t$  line (bottom of chart, Fig. 11) will give the same efficiency ratio as the (5'-7) line (top of chart, Fig. 11). The points so located show that (4') and (6')

$$\eta_m/\eta_t = 0.835$$

and

$$\tan \phi = W/M = 0.875^*$$

In order to keep the rate of water circulated constant, it will be necessary to increase the air flow rate by  $(1.1-0.875)/0.875 = 0.257$  or 25.7 percent above the original rate. This will be simpler and more economical to operate under changing weather conditions. Instead of increasing the pump and fan capacities by 15 percent, simply increase the air flow rate by 25.7 percent.††

#### REFERENCES

1. *Air Conditioning Analysis*, by William Goodman (The Macmillan Co., New York, N. Y., 1943).
2. *Heat Transmission*, by W. H. McAdams (McGraw-Hill Book Co. Inc., New York, N. Y., 1942, p. 228).
3. Performance of a Forced-Draft Cooling Tower, by B. H. Spurlock, Jr. (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 311).

#### APPENDIX

##### MATHEMATICS OF THE COOLING TOWER PROCESS

Equation 4 which fixes the slope of the operating line is developed mathematically by the following analysis of what takes place during the cooling tower process.

All that is known at the start are the water temperatures ( $t_2, t_1$ ) at the top and bottom of the tower, respectively, and the dry- and wet-bulb temperatures fixing the enthalpy of the air entering at the bottom of the tower. If  $W$  and  $M$  are the flow rates of the water and air, respectively, in pounds per min, the water at the top of the tower will have a total enthalpy

$$I_w = W \times i_w = W(C_{pl})(t_2 - t_0) \dots \dots \dots (A-1)$$

and the air-vapor mixture leaving the tower in the same plane will have a total enthalpy

$$I_m = M \times i_m \dots \dots \dots (A-2)$$

At a plane section, a small distance below the entrance of the water at the top of the tower, a small amount of water  $\Delta W$  will have evaporated and thereby lowered the temperature  $t_2$  of the remaining  $(W-\Delta W)$  by the small temperature decrease  $\Delta t$ . This will have effected a rise in the enthalpy of the air. At this section, therefore, the enthalpy of the air will be less than the enthalpy at the top of the tower by the amount  $\Delta i_m$ . Analogous to the foregoing, the respective total enthalpies for the plane section can be expressed by

\* It will not always be possible to locate common values on the  $\eta_m/\eta_t$  scale and in the grid. This indicates that the tower will not give the required performance without changing the cooling range and the amount of water circulated.

†† The example demonstrates the influence of changing wet-bulb temperature on the performance of the tower for a constant inlet water temperature. This should not be confused with the case shown in Fig. 10, where the temperature of the inlet water varies in the same amount as the wet-bulb temperature. The results obtained in the former case are exactly opposite to those in the latter, which corresponds to that of different barometric pressures.

$$I'_w = W' C_{pl}[(t_a - \Delta t) - t_o] \quad (A-3)$$

and

$$I'_m = M(i_m - \Delta i_m) \quad (A-4)$$

where

$$W' = W - \Delta W$$

Assuming that the tower is operated with a heat loss relatively low in comparison to the heat exchange involved, the increase in the air enthalpy between the 2 cross-sections must equal the decrease in the enthalpy of the water between the same 2 cross-sections.

Subtracting Equations A-3 and A-4 from Equations A-1 and A-2 the following heat balance is obtained

$$I_w - I'_w = I_m - I'_m \quad (A-5)$$

$$W C_{pl}(t_2 - t_o) - W' C_{pl}[(t_2 - \Delta t) - t_o] = M i_m - M(i_m - \Delta i_m) \quad (A-6)$$

The amount of evaporation over the whole tower is quite small for practical purposes and may be neglected. Hence

$$W = W'$$

and equation A-6 reduces to

$$W C_p(\Delta t) = M(\Delta i_m) \quad (A-7)$$

The specific heat of the water ( $C_{pl}$ ) may be taken as unity in this range, which gives for Equation A-7 the relation

$$\frac{W}{M} = \frac{\Delta i_m}{C_{pl}(\Delta t)} = \frac{\Delta i_m}{\Delta t} \quad (A-8)$$

which is the basis of the following development (see Reference 2 of paper, pp. 285-292).

Since the water-air ratio, once selected, is practically constant for the process, the quantity  $\frac{\Delta i_m}{\Delta t}$  must be constant over the whole cooling range, that is, from the top of the tower to the bottom. This establishes the slope ( $\tan \phi$ ) of a straight line, the so-called *operating line*; and because the enthalpy of the air  $i_{mA}$  and the temperature of the water  $t_1$ , both at the bottom of the tower are fixed and known, the line ( $\tan \phi$ ) must pass through the point (A') representing the initial condition of the air and the final condition of the water. That is to say,

$$\tan \phi = \Delta i_m / \Delta t = \frac{W}{M}, \text{ which is Equation 4.}$$

In this connection it is interesting to note that this process (operating line) on the enthalpy chart is a straight line while the enthalpy *progress curve* is not. (Fig. 5, line A'-B', and Fig. 3, line a-b, respectively). This indicates that the heat transfer between the water and the air is a linear function, whereas the actual increase in the enthalpy of the air or the decrease in the water is not.

Once the position of the operating line is defined and the air conditions are known, the path of the conditioning curve as well as the progress curve is completely determined. The slope and the position of the operating line (Fig. 5, A'-B') cannot be selected arbitrarily. As explained, the point (A) is fixed by the final water temperature  $t_1$  and the enthalpy of the entering air  $i_{mA}$ , which in turn, is a function of the wet-bulb temperature. The terminal point (B) is defined by the temperature  $t_2$  of the water at the top of the tower and the slope of the operating line ( $\tan \phi$ ).

#### COMMENTS ON EQUATION 5

Equation 5 is of course a practical means of defining the efficiency of the cooling tower and has no theoretical basis. In reality the efficiency of the tower should be ex-



To develop and explain this method consider the area A'-B'-F-E in Fig. 5 reproduced to a larger scale in Fig. A-1. As a first step divide this area into  $p$  vertical strips of equal width, (see Fig. A-1),  $(t_2 - t_1)/P$  where  $(t_2 - t_1)$  is the cooling range.

The integral in Equation 8 may be replaced by the term

$$N \approx \sum_{i_{mA}}^{i_{mB}} \frac{\Delta i_m}{(i_w - i_m)} \approx \sum_{i_{mA}}^{i_{mB}} \frac{(i_{mB} - i_{mA})}{P} \times \frac{1}{(i_w - i_m)}$$

$$N = \frac{(i_{mB} - i_{mA})}{P} \times \sum_{i=1}^P \frac{1}{\delta} \quad \dots \dots \dots (A-9)$$

where

$$\delta = (i_w - i_m)$$

is the instantaneous overall driving force of each individual increment.

If  $\delta_m$  is the mean driving force

$$\delta_m = (i_w - i_m), \text{ (subscript } m \text{ indicates mean)}$$

over the entire range (A') to (B'), then

$$\frac{1}{\delta_m} = \frac{1}{P} \left| \frac{1}{\delta_1} + \frac{1}{\delta_2} + \dots + \frac{1}{\delta_P} \right| = \frac{1}{P} \sum_{i=1}^P \frac{1}{\delta} \quad \dots \dots \dots (A-10)$$

Substituting

$$\frac{(i_{mB} - i_{mA})}{t_2 - t_1} = \tan \phi,$$

and Equation (A-10) into equation (A-9),

$$N = \frac{(t_2 - t_1) \tan \phi}{\delta_m} \quad \dots \dots \dots (A-11)$$

If the cooling range is not too great, one\* may express  $\delta_m$  as the logarithmic mean between the driving force  $\delta_a$  and  $\delta_b$  at the points (A') and (B'), respectively, where

$$\delta_{\log \text{ mean}} = \frac{\delta_b - \delta_a}{\ln \frac{\delta_b}{\delta_a}} \quad \dots \dots \dots (A-12)$$

The following problem illustrates the way Equations (A-10) and (A-11) can be used to solve for  $N$ , and how the solution is affected by using Equation (A-12) instead of (A-11) for determining the value of  $\delta$ .

*Problem: Determine the effective height and the gross area for the same cooling tower, whose design data was given under the heading "Cooling Tower Performance." The average air velocity shall not be less than 250 fpm, measured across the total tower area.*

*Solution:* From previous data,  $M = 1200$  lb per min, or 16,000 cfm standard air;  $W = 120$  gpm or 1,000 lb per min; cooling range 15 F,  $\tan \phi$  is 0.835.

The required tower area is then

$$S = 16,000 \text{ cfm} \times \frac{1}{250 \text{ ft/min}} = 64 \text{ sq ft.}$$

The mass air velocity is

$$M = \frac{1200}{64} = 18.7 \text{ lb per (min) (sq ft)}$$

\*HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1947, p. 669 (published by AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.)

and the mass water velocity is

$$W = \frac{1000}{64} = 15.6 \text{ lb per (min) (sq ft).}$$

Examining Table 1, the value of  $Z_{og}$  in the order of the above mass velocities, is found to be

$$Z_{og} = 19 \text{ ft}$$

for a grid packing. Divide the area of the driving forces into ( $P = 10$ ) equal parts (see Fig. A-1) and sum up the reciprocal values of  $\delta$  to find

$$\begin{aligned} \frac{1}{\delta_m} &= \frac{1}{4.00} + \frac{1}{4.13} + \frac{1}{4.25} + \frac{1}{4.37} + \frac{1}{4.55} + \frac{1}{4.75} + \frac{1}{5.00} + \frac{1}{5.30} + \frac{1}{5.80} + \frac{1}{6.25} \\ &= \frac{1}{4.75} \text{ Btu} \end{aligned}$$

$$N = \frac{(t_2 - t_1) \tan \phi}{\delta_m}, \text{ which is Equation A-11.}$$

$$= \frac{15 \times 0.835}{4.75} = 2.64,$$

and the effective tower height is

$$Z = N Z_{og} = 2.64 \times 19 = 50 \text{ ft.}$$

Using Equation A-12 to find the log mean value of  $\delta_m$ , one finds  $\delta_{\log \text{ mean}} = 5.1$  and

$$N_{\log \text{ mean}} = \frac{15 \times 0.835}{5.1} = 2.45$$

a value 7 percent below the 2.64 obtained by use of (A-11).

#### DEVELOPMENT AND USE OF EQUATION 9

Graphical integration of Equation 8 is thus seen to be elaborate and cumbersome. A simpler and faster method is needed and one which eliminates a detailed graphical analysis of the psychrometric chart.

The difficulties of graphical integration may be reduced but not eliminated by using the *log mean* value of  $\delta_m$  obtained by Equation (A-12), but to do this one must still evaluate the values of  $\delta_a$  and  $\delta_b$ , the end values. Litchenstein† points out that this method has to be handled with care as erroneous results may occur.

In order to gain a new approach to the problem, refer to Equations (A-10) and (A-11). It is evident that the mean value  $\delta_m$  can be approximated by the term

$$\frac{1}{\delta_m} = \sum_{a}^b \frac{1}{\delta} \approx \frac{1}{\delta_a} + \frac{1}{\delta_b} \dots \dots \dots \text{ (A-13)}$$

The value  $\delta_b$ , represented by the distance (F-B') in Figs. 5, 6 and 8, can be defined in terms of known values. Analyzing the geometrical relations in Fig. 7, it is seen that

$$\frac{1}{\delta_b} = \frac{\eta_m}{1 - \eta_m} \times \frac{1}{(t_2 - t_1) \tan \phi} \dots \dots \dots \text{ (A-14)}$$

The quantity  $\delta_a$  is not as easily obtained. Furthermore, in order to compensate for the curvature of the enthalpy curve, the value of  $\delta_a$  to be used in equation (A-13) will

† Performance and Selection of Mechanical Draft Cooling Towers, by Jos. Litchenstein (*ASME Transactions* October 1943, pp. 779-88).

have to be smaller by  $(\delta_a - \delta_{a'})$ , which results in a value of

$$\delta_{a'} = H - A'$$

in Fig. 7 and which is related to

$$\delta_a = E - A' \text{ by a factor } k.$$

One may express

$$\frac{\delta_a}{\delta_{a'}} = \frac{E - A'}{H - A'} = k \quad \text{. . . . . (A-15)}$$

and by means of Fig. 7, it will be found that

$$\frac{K - A'}{E - A'} = \frac{K - A'}{L - A'} \cdot \frac{L - A'}{E - A'} = \frac{\tan \psi}{\tan \alpha} \cdot \frac{\tan \phi}{\tan \psi} \quad \text{. . . . . (A-16)}$$

where  $\psi$  and  $\alpha$  represent the angles of the chords between the terminal points ( $a''$ ) and (F), and ( $a''$ ) and (E), respectively. With  $\tan \phi$  already known, the following geometrical relations will be found to exist.

$$\frac{\tan \phi}{\tan \psi} = \frac{\eta_m}{\eta_t} \quad \text{. . . . . (A-17)}$$

and

$$K - A' = \left( \frac{1 - \eta_t}{\eta_t} \right) (\tan \phi) (t_2 - t_1) \quad \text{. . . . . (A-18)}$$

By means of Equations A-15, A-16, A-17 then

$$\frac{1}{\delta_{a'}} = \frac{1}{H - A'} = k \left( \frac{\tan \psi}{\tan \alpha} \right) \left( \frac{\eta_m}{1 - \eta_t} \right) \cdot \left[ \frac{1}{\tan \phi (t_2 - t_1)} \right] \quad \text{. . . . . (A-19)}$$

Since  $\delta_a$  is greater than  $\delta_{a'}$ , the quantity  $k$  is greater than unity and since the  $\tan \psi$  is greater than  $\tan \alpha$ , the ratio

$$\frac{\tan \psi}{\tan \alpha}$$

is greater than unity and

$$k \left( \frac{\tan \psi}{\tan \alpha} \right)$$

is also greater than unity. Therefore, it can be written

$$\begin{aligned} \frac{k \tan \psi}{\tan \alpha} &= (1 + g), \text{ and} \\ \frac{1}{\delta_{a'}} &= (1 + g) \left( \frac{\eta_m}{1 - \eta_t} \right) \left[ \frac{1}{\tan \phi (t_2 - t_1)} \right] \quad \text{. . . . . (A-20)} \end{aligned}$$

Adding Equations A-14 and A-20 one has the equivalent expression for Equation A-10, which upon substitution into Equation A-11 gives

$$N = \frac{1}{2} \left[ \frac{\eta_m}{1 - \eta_t} + \frac{\eta_m}{1 - \eta_m} \right] + \frac{g}{2} \left( \frac{\eta_m}{1 - \eta_t} \right) \quad \text{. . . . . (A-21)}$$

The only term so far not accounted for on the right side of this equation is the factor  $g$ , which represents the influence of the curvature of the enthalpy curve on the final result.<sup>‡</sup>

<sup>‡</sup> This influence may be negligible in the case of small temperature ranges and/or low efficiencies, but under different circumstances it can be appreciable. Expressed in other words, this means that the larger the cooling range, and/or the higher the efficiencies, and/or the greater the water-to-air ratio, the more pronounced is the effect of the curvature on the size of the area of the driving forces (Fig. A-1 and 7), the smaller will be therefore the resulting mean value  $\delta_m$  and the greater consequently the number  $N$ .

An exact derivation of  $q$  in consistent terms would be by coincidence. But for the purpose of this paper it is not required, because the last term of Equation A-21 will rarely exceed 25 percent of the total results. Over the usual operating ranges  $q$  will approximately amount to

$$q = 0.24 \frac{\eta_t}{1 - \eta_m} \tan \phi.$$

Hence, equation (A-21) can be written as

$$N = \frac{\eta_m}{2} \left[ \frac{1}{1 - \eta_t} + \frac{1}{1 - \eta_m} \right] + \frac{0.12 \eta_m \eta_t \tan \phi}{(1 - \eta_m)(1 - \eta_t)}, \text{ which is Equation 9.}$$

an expression for the number of *transfer units* that contains only known or assumed design values and these values may be calculated with pencil and slide rule only. Within the general operating limits this equation gives results within a tolerance of  $\pm 10$  percent.

With the use of this equation, the number  $N$  of transfer units, computed in the previous problem amounts to

$$\begin{aligned} N &= \frac{0.65}{2} \left[ \frac{1}{1 - 0.75} + \frac{1}{1 - 0.65} \right] + \frac{0.65 \times 0.75 \times 0.12 \times 0.835}{(1 - 0.75)(1 - 0.65)} \\ &= 2.23 + 0.56 = 2.78, \end{aligned}$$

which differs by only 5 percent from the result (2.64) obtained by graphical integration of Equation 8.

A *rule of thumb* method that one might use is

$$N \approx 0.6 \eta_m \left[ \frac{1}{1 - \eta_t} + \frac{1}{1 - \eta_m} \right]. \quad \dots \quad (\text{A-22})$$

Equations 9 and A-22 show very clearly the tendency of tower height to approach infinity with increasing efficiencies as was pointed out earlier.

#### DEVELOPMENT OF EQUATION 10

In order to express  $\tan \phi$  in Equation 9 in terms of known basic data, the following procedure is used.

Begin by considering Equation A-17, and write it in the form:

$$\tan \phi = \frac{\eta_m}{\eta_t} \tan \psi$$

In order to define  $\tan \psi$ , the enthalpy must be expressed in mathematical form. The equation

$$i = e^{\left(\frac{T}{40} + 1.75\right)} \quad \dots \quad (\text{A-23})$$

follows the enthalpy curve with sufficient accuracy, at sea level pressure.

Since the slope of a chord between any two points  $x$  and  $y$  on the enthalpy curve is given by

$$\tan \psi_{xy} = \frac{i_x - i_y}{T_x - T_y} \quad \dots \quad (\text{A-24})$$

it follows from Equation A-17, that

$$\begin{aligned} \tan \phi &= \left( \frac{\eta_m}{\eta_t} \right) \left[ \frac{(i_F - i_{a'})}{(t_2 - T_{WB})} \right] \\ &= \frac{\eta_m}{\eta_t} \times e^{\left(\frac{T_{WB}}{40} + 1.75\right)} \times \left[ \frac{e^{\left(\frac{t_2 - T_{WB}}{40}\right)} - 1}{t_2 - T_{WB}} \right] \text{ which is Equation 10.} \end{aligned}$$



**1552**

## RESISTANCE OF WOODEN LOUVERS TO FLUID FLOW†

By C. W. BEVIER\*, COLLEGE STATION, TEX.

THERE HAS been engineering thought to the effect that louvers<sup>1</sup> used for air intakes on cooling towers may be responsible for as much as 20 percent of the total pressure loss of induced draft cooling towers and that spray eliminators (another arrangement of louvers) may account for an additional 20 percent.<sup>2</sup> Also of concern have been the louvers used for both intake and outlet of some attic fan installations in which the pressure drop affects both capacity and cost of operation.

It is obvious from these examples that any change in the design of louvers which reduces the cost of operation without adversely affecting other costs is desirable. The research reported here was undertaken with such considerations in view and was concerned primarily with the measurement of pressure drop through louvers of several designs to determine not only the characteristics of those louvers, but also principles of general application.

The general approach to this subject and an analysis of the problem together with an explanation of how the calculations were made are all contained in the Appendix.

### EQUIPMENT

The test setup illustrated in Figs. 1 and 2 consists of a 24-in. centrifugal fan (driven by a 2-hp, 3-phase, 220-volt motor) discharging air through one of two 13-in. diameter by 11 ft long ducts into a plenum chamber 7 ft by 11 ft in plan dimensions and 7 ft high.

Attached to the exit of the plenum chamber is a square duct 4.5 ft long and 3 sq ft in area. The louver to be tested is fixed to the end of this duct. Inside the plenum chamber are 5 concentric cone diffusers designed to distribute the air uniformly and to give the chamber the effect of a large room. In addition, there is a bell-shaped entrance leading from the plenum chamber into the 3 sq ft duct extension, the purpose of which was to direct the air smoothly to the inlet side of the louver.

One of the 13-in. ducts connecting the fan and chamber contains a 6-in. diameter flow nozzle used to measure low-to-intermediate rates of flow. The other duct is equipped for a 20-point Pitot tube traverse used to measure intermediate-to-high rates of flow. The ducts are so arranged that they can be lowered or raised

† Based on the author's thesis submitted to the Graduate School of A. & M. College of Texas in fulfillment of a requirement for an M.S. in Mechanical Engineering.

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<sup>1</sup> Exponent numerals refer to References.

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into position as the air flow rate requires. For a given test the air flow rate was fixed by the adjustment of movable vanes at the entrance to the fan.

Pressure measurements were made with inclined and inclined-vertical manometers. The manometers were calibrated with a micro-manometer, and all read-

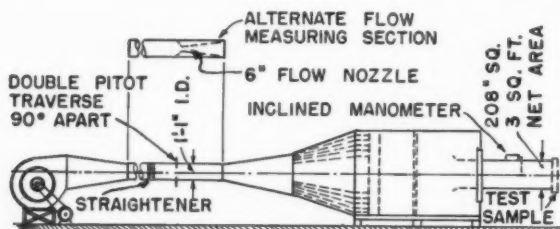


FIG. 1. TEST SETUP

ings corrected for temperature to standard density of water. The velocity pressure for the flow nozzle was obtained from a manometer connected between an upstream static tap and a tap at the throat of the nozzle. The same manometer was connected to the Pitot tube when it was in use. The static pressure loss through the louver was obtained from a manometer connected to a static pressure tap located



FIG. 2. LOUVER AND INLET DUCT ATTACHED TO PLENUM CHAMBER

2 ft in front of the louver within the duct extension to which the louver was connected. Barometric pressure was determined from a mercury column barometer and corrected for temperature, elevation, and gravity by usual methods.

Wet- and dry-bulb temperature measurements were made with two 0- to 120 F mercury-filled glass thermometers having 0.5 deg graduations. The thermometers were placed near the inlet side of the blower.

## DESCRIPTION OF LOUVERS

Tests were run on 4 types of louvers as illustrated in Fig. 3. All louver boards were 20.78 in. long; made from wood stock  $\frac{13}{16}$  in. thick, and sanded to a smooth finish so that the surface conditions were nearly uniform throughout all tests.

Louvers were assembled by using sheet metal flanges as shown in Fig. 2. Various arrangements for each type of louver were obtained by changing the width, the number of louver boards or angle of slope.

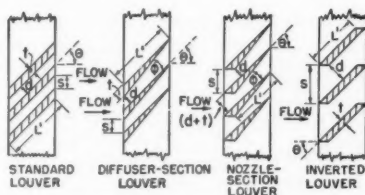


FIG. 3. LOUVER ARRANGEMENTS TESTED

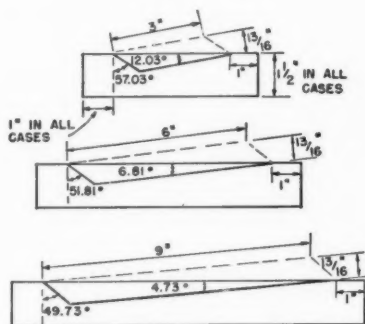


FIG. 4. LOUVER BOARD JIGS

The first series of tests was run on the standard louvers. These louvers were made of boards of uniform thickness and widths of 3, 6,  $8\frac{1}{2}$ , and 9 in.

Results obtained from these tests indicated that the width of the louver boards has an appreciable effect on the pressure loss; that is, the exit portion of the louver affects the flow.

This observation was the reason for testing the diffuser-section louvers. It was believed that if a diffuser-type exit section could be incorporated in the louver, a pressure recovery would occur similar to that obtained in the diffuser section of a venturi tube, ejector, or similar device. Boards for the diffuser-section louvers were made from the standard louver boards by placing them in a jig and passing the assembly through a planer which cut off the upper half of the boards, thus leaving the desired shape. Dimensions of the 3 sizes of diffuser-section louvers tested are shown in Fig. 4.

In testing the diffuser-section louvers, observations led to the belief that turbulence caused by the type of entrance was appreciably affecting the flow conditions and possibly accounted for the increase in pressure loss as compared to the standard louver. With this thought in mind, it was decided to invert the diffuser-section louver boards and thus obtain a shape approximating that of an airfoil. The flow passage thus formed was similar to that of a nozzle or venturi and, consequently, these louvers were called nozzle-section louvers.

The considerations which led to the inversion of the diffuser-section louver boards also suggested the inversion of the standard louver boards. To simplify the geometry of the inverted louvers thereby obtained, a zero overlap of the louver boards was maintained throughout this series of tests. The different arrange-

TABLE 1—SUMMARY OF LOUVER ARRANGEMENTS TESTED

Standard Louvers			
Width of Louver Boards, in.	Number of Louver Boards	Approach Angle, $\theta$ , Deg	
3	6, 7, 8, 9	45	
6	4, 5, 6, 7, 8, 9	45	
$8\frac{1}{8}$	7, 8, 9	45	
9	4, 5, 6, 7, 8, 9	45	

Diffuser-Section Louvers			
Width of Louver Boards, in.	Number of Louver Boards	Approach Angle, $\theta$ , Deg	Diffuser Angle, $\phi$ , Deg
3	8	45	12.03
6	8	45	6.81
9	8	45	4.73

Nozzle-Section Louvers			
Width of Louver Boards, in.	Number of Louver Boards	Approach Angle, $\theta$ , Deg	Diffuser Angle, $\phi$ , Deg
6	4, 5, 6, 7, 8	45	6.81
9	4, 5, 6, 7, 8	45	4.73

Inverted Louvers		
Width of Louver Boards, in.	Number of Louver Boards	Approach Angle, $\theta$ , Deg
6.00	4	30
4.00	6	30
3.00	8	30
7.35	4	45
4.90	6	45
3.68	8	45
10.39	4	60
6.93	6	60
5.20	8	60

ments of louver boards were obtained by varying the spacing of the louver boards, which, because of maintaining zero overlap, required changing the widths of the boards.

The inverted louver boards were made for approach angles of 30, 45, and 60 deg. The widths varied from 2.99 in. to 10.39 in.

### EXPERIMENTATION

The test procedure consisted of selecting a particular type of louver board, varying the dimensions and arrangement of the louver boards, and of obtaining the pressure loss through the louver at different rates of air flow. Because the face area of the louvers was fixed by the size of the entrance duct, the values of  $d$ ,  $s$ , and the free area ratio were determined by the number of boards used in any given arrangement. Table 1 gives a summary of the various louver arrangements tested. Most of the louvers were tested at 4 different rates of flow adjusted to give approximately equal increments of pressure loss.

Air volumes were measured by means of either the 6-in. flow nozzle or the Pitot tube. The flow nozzle was used whenever possible because of the ease of obtaining data. However, as the free area ratio increased, the air volume increased beyond the capacity of the nozzle; therefore, fewer tests were possible using this method, and the Pitot tube was then used. To check the difference in results obtained by the 2 methods 2 tests were run completely using the nozzle and then rerun using the Pitot tube. The difference in the results from the 2 methods was negligible.

In all tests, flow and temperatures in the system were allowed to become steady before any measurements were made. When using the flow nozzle 3 readings of the manometers were taken during each run, while temperatures were recorded only at the beginning and end of a run. The average of the readings was used in calculations.

When the Pitot tube was used, the velocity pressure was obtained by squaring the average of the square roots of 20 traverse points located at the centers of equal annular areas as prescribed by the fan test code.<sup>3</sup> Manometer readings for the pressure drop through the louvers, and the temperatures, were recorded at the beginning, halfway point, and end of the traverse and the average of the readings was used in calculations.

At all times care was taken to insure a minimum amount of air leakage between the point of flow measurement and the louver. Masking tape was used to seal cracks. The test setup was checked for leakage and the amount found to be less than 1 percent of the air delivered by the fan, using the test for leakage previously outlined.<sup>4</sup>

The velocity front of the air in the 13-in. flow measurement duct containing the Pitot tube was checked for various positions of an egg-crate type straightener. The straightener was finally located at the entrance to the duct, because this location gave the most uniform velocity front at the Pitot tube.

### RESULTS AND DISCUSSION

*Standard Louvers:* Fig. 5 represents graphically the effect of the width of the louver boards on louver performance. Because the curves are for a given pseudo Reynolds number, and because ratio  $d$  is constant for a given free area, each curve of Fig. 5 approximately represents a constant velocity depending on the magnitude

of the free area ratio. Under these conditions a decrease in the resistance coefficient for a given free area ratio gives a decrease in the pressure loss through the louver. These curves indicate, therefore, that for a given area ratio, as the width

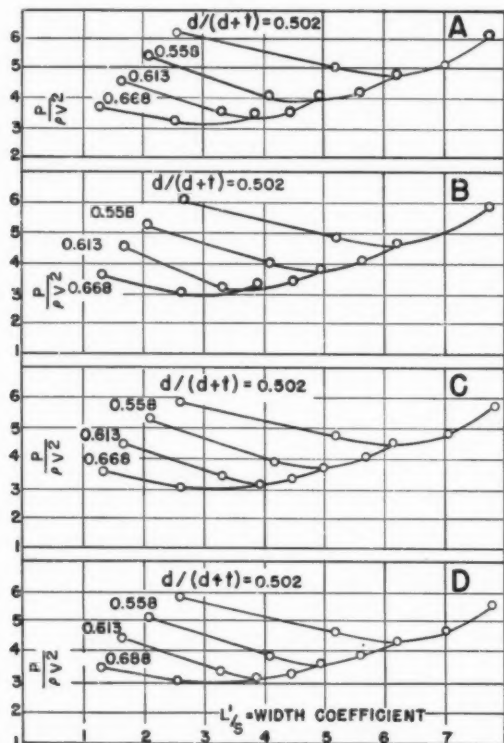


FIG. 5. VARIATION OF RESISTANCE COEFFICIENT ( $P/\rho v^2$ ) WITH THE WIDTH COEFFICIENT ( $L'/s$ ) FOR DIFFERENT VALUES OF FREE AREA RATIO. PSEUDO REYNOLDS NUMBER IS CONSTANT AT 4000 AT A, 6000 AT B, 8000 AT C, AND 10,000 AT D.  $d/s = \cos \theta = 0.707$ , ALL CASES (STANDARD LOUVERS)

of the louver board increases, the pressure loss through the louver decreases until a minimum is reached; then further increase of the width causes the pressure loss to increase.

Apparently, as the width increased, the flow conditions were at first improved and a smaller turbulence loss occurred, resulting in a smaller overall pressure loss. However, as the width was further increased, the friction loss increased with enlarged surface area, thereby causing greater overall resistance.

The curves in Fig. 5 also show that, as the free area ratio increases, the minimum value of the resistance coefficient occurs at smaller values of the width coefficient. And, if the thickness is held constant, it can be shown by reference to

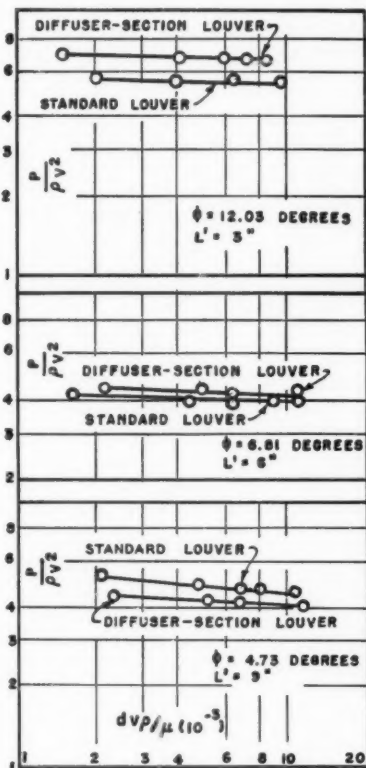


FIG. 6. VARIATION OF RESISTANCE COEFFICIENT  $(P/dv^2)$  WITH REYNOLDS NUMBER  $[dv\rho/\mu(10^{-3})]$  FOR DIFFERENT VALUES OF FREE AREA RATIO (DIFFUSER-SECTION LOUVER AND STANDARD LOUVER).  $\theta = 45$  DEG, ALL CASES.  $d/(d+t) = 0.558$

these curves, that a decrease in the free area ratio causes a decrease in the width required for minimum resistance. It may also be shown that a decrease in the thickness causes a very noticeable decrease in the width required for minimum resistance. These effects can be explained by the fact that as the thickness is

reduced for a given free area ratio, the area open to flow is reduced and the velocity in the louver must increase. The increase in velocity causes the friction loss to increase, and hence an overall increase in pressure loss occurs at a smaller width.

As the free area ratio increases, the change in the resistance coefficient becomes progressively less. Because the velocity within the louver is reduced as the free area ratio is increased, a decrease in the turbulence results. Hence, additional reduction in the turbulence with increasing width of the louver board may not follow. Also, the fluid friction is reduced for higher free area ratios, because there

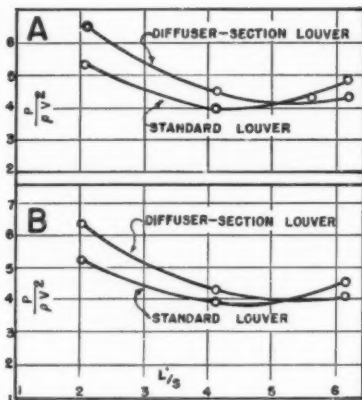


FIG. 7. VARIATION OF RESISTANCE COEFFICIENT ( $P/\rho v^2$ ) WITH THE WIDTH COEFFICIENT ( $L'/s$ ) FOR DIFFUSER-SECTION LOUVERS AND STANDARD LOUVERS. PSEUDO REYNOLDS NUMBER [ $dv\rho/\mu(10^{-3})$ ] IS 4000 AT A, AND 8000 AT B.  $d/(d+t) = 0.558$ , ALL CASES.  $\theta = 45$  DEG, ALL CASES

are fewer boards per unit of face area, and a corresponding reduction of surface area. Consequently, the change in the friction loss with a change in width becomes less significant.

If the curves of Fig. 5 were superimposed, it would be seen that the curves almost coincide. This means that the pseudo Reynolds number affects these curves very little. Evidently the flow through the louvers is so turbulent that the viscous forces are not significant. This conclusion is strengthened by the fact that at the low pseudo Reynolds numbers the curves tend to be less coincident, which is consistent with the fact that at low Reynolds numbers the flow becomes less turbulent and approaches laminar flow conditions.

**Diffuser-Section Louvers:** Results of tests of the diffuser-section louvers are shown in Figs. 6, 7 and 8. In these figures a comparison is made between the diffuser-section louver and the standard louver, each having the same free area ratio, width, and approach angle.

The change in the resistance coefficient with a change in the pseudo Reynolds number for a given width, free area ratio, and approach angle is shown in Fig. 6. These curves show that as the width of the boards increases, the resistance of the diffuser-section louver decreases, and that, compared to the standard louver, for a louver board width of 3 in. the resistance of the diffuser-section louver is greater, but for a 9-in. width it is less than that of the standard louver. This result can be explained by the fact that both the diffuser angle and the discharge angle of the air are reduced as the width is increased.

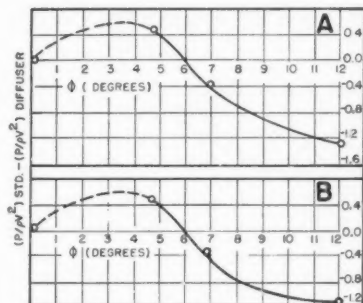


FIG. 8. VARIATION OF DIFFERENCE BETWEEN RESISTANCE COEFFICIENTS FOR STANDARD LOUVER AND DIFFUSER-SECTION LOUVER WITH DIFFUSER ANGLE. PSEUDO REYNOLDS NUMBER  $[dvp/\mu(10^{-3})] = 4000$  AT A, AND 8000 AT B.  $\theta = 45$  DEG, ALL CASES

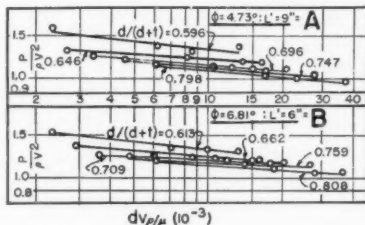


FIG. 9. VARIATION OF RESISTANCE COEFFICIENTS ( $P/P_{STD}$ ) WITH PSEUDO REYNOLDS NUMBER  $[dvp/\mu(10^{-3})]$  FOR DIFFERENT VALUES OF FREE AREA RATIO. NOZZLE-SECTION LOUVERS WITH  $\theta = 45$  DEG, ALL CASES

If the resistance coefficient is plotted against the width coefficient for a given pseudo Reynolds number and free area ratio as in Fig. 7, the minimum resistance for the standard louver is less than that for the diffuser-section louver. However, at high width coefficients the diffuser-section louver has less resistance than that of the standard louver.

The effect of the diffuser angle on louver performance is shown in Fig. 8.

**Nozzle-Section Louvers:** The results of tests on the nozzle-section louvers are shown in Fig. 9. The curves indicate that the width and free area ratio of nozzle-section louvers do not appreciably affect the resistance coefficient. Apparently the flow conditions are improved owing to the shape of the entrance section by an amount such that only a minor recovery occurs by adding to the width or spacing of the boards.

It is seen by comparing the results of the standard louver and nozzle-section louver tests that the resistance coefficient for a nozzle-section louver of a given free area ratio and width of louver board is approximately one-half that of the standard louver.

**Inverted Louvers:** The curves of Fig. 10 show the performance characteristics of inverted louvers having zero overlap of the louver boards.

The figure contains 3 major groups of curves, each for a different value of  $s/L'$  corresponding to slope angles of 30, 45 and 60 deg. Within each group individual curves correspond to different constant values of free area ratio and also different board widths. Changing the board width with changes in spacing or angle of slope was necessary in order to maintain zero overlap.

These curves show that as  $s/L'$  increases, that is, as the approach angle increases, the free area ratio becomes more significant as indicated by the wider spread of the resistance coefficient. Apparently, as the angle increases, the turbulence immediately downstream from the entrance edge of the louver board increases, but the overall resistance is diminished by increasing the spacing and also the width of the louver board, thus accounting for the greater significance of the free area ratio revealed by the curves.

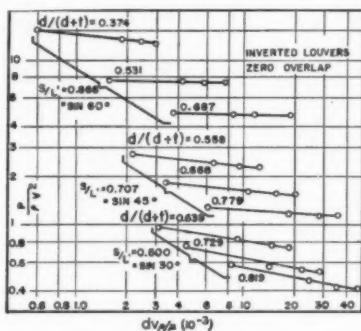


FIG. 10. VARIATION OF RESISTANCE COEFFICIENT WITH PSEUDO REYNOLDS NUMBER FOR DIFFERENT VALUES OF FREE AREA RATIO AND  $s/L'$  (INVERTED LOUVERS)

If the 45-deg inverted louver is compared with the 45-deg standard louver of the same free area ratio and approximately the same width, it is seen that the resistance coefficient of the inverted louver is about 40 percent less than that of the standard louver. These louvers were made from exactly the same boards, the only difference being that in one case the boards were assembled with their beveled edges in a vertical plane, whereas in the second case (inverted louvers) the beveled edges were horizontal, that is, parallel to the direction of air flow. The major difference here is that the inverted louvers were arranged to give better entrance conditions.

In view of the better performance of the inverted louver, it seems that this type louver should be more desirable in installations where pressure loss is a decisive factor.

The results of the inverted louver investigation are presented here in graph form, using as parameters the several dimensionless coefficients, as explained in the Appendix, in order to make them apply generally to a wide variety of applications. Other coordinates might have been used, but with the loss of generality. For

example, in Fig. 11 pressure drop in inches of water for the inverted louver of 45 deg slope is plotted against cfm per sq ft of face area for 3 different board widths in inches. The equation of the family of curves was found to be

$$P = [1.512 + 152.5(L')^{-2.672}] 10^{-7} v^{1.836 + 0.0128 L'}$$

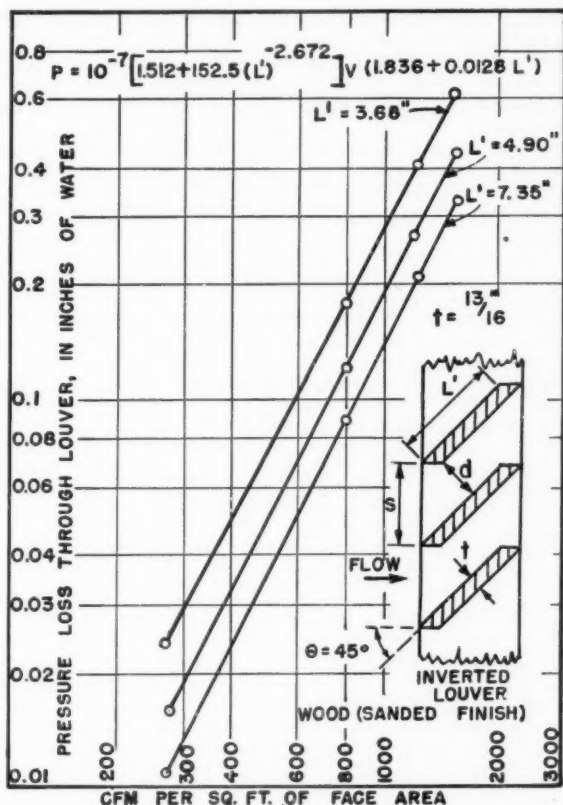


FIG. 11. VARIATION OF PRESSURE LOSS THROUGH LOUVER WITH FACE VELOCITY FOR DIFFERENT VALUES OF THE WIDTH

The range of values of  $L'$  and  $v$  were 3.68–7.35 in. and 250–2000 cfm respectively. The equation should not be used for values of the variables very much outside these ranges. Also, neither the curves nor the equation take into consideration variations in air density or viscosity. For the tests from which this equation was derived, the range in air density was 0.07055–0.7231 lb<sub>m</sub>/ft<sup>3</sup> and in viscosity it was

124.1–126.1 lb<sub>m</sub>/ft-sec. However, for many louver applications the density and viscosity of the air may not differ appreciably from these values, and the equation may apply. For applications where the above assumptions are not valid the curves of Fig. 10 should be used.

#### REFERENCES

1. Pressure Loss of Air Flowing through 45-Degree Wooden Louvers, by P. R. Cobb ASHVE TRANSACTIONS, Vol. 60, p. 43).
2. Private communication with W. W. Smith, Texas Engineering Experiment Station, A. & M. College of Texas. The statement was based on model studies conducted by Professor Smith for the Texas Engineering Experiment Station in connection with cooling tower research.
3. Standard Test Code for Centrifugal and Axial Fans (*National Association of Fan Manufacturers, Inc.*, Bulletin No. 110, Second Edition, 1952).
4. Determination of Air Infiltration in Fan Testing Equipment, by W. D. Scoates (Research Report No. 40, Texas Engineering Experiment Station, A. & M. College of Texas).

#### APPENDIX

The current formulation of fluid resistance is based on a modification of Newton's equation and may be represented by  $D = \text{Drag} = f(R) A \rho v^2$ , where  $f(R)$  is a function of Reynolds number and varies for each shape and position of the body. This equation thus takes into consideration pressure differences, frictional stresses within the fluid, and the resistance, drag, or frictional effect of the fluid flowing around the body. Therefore,  $f(R)$  must be determined by experiment.

Specifically, it was attempted to analyze various shapes and arrangements of louver boards according to the principles of similarity and dimensional analysis, to find by experiment the value of the various factors or dimensionless groups which characterize a given arrangement, and to present the results in a manner useful for the solution of practical design problems.

#### ANALYSIS OF PROBLEM

Variables considered in applying the method of dimensional analysis to the louvers used in this investigation are as follows:

- $P$  = pressure loss through the louver, inches of water.
- $v$  = velocity of the air in the approach duct (cfm/sq ft of face area of the louver), feet per minute.
- $\rho$  = density of the air, pounds mass per cubic foot.
- $\mu$  = viscosity of the air, pounds mass per foot per second.
- $d$  = perpendicular distance between louver boards, inches.
- $s$  = distance between louver boards at entrance face of louver, inches.
- $t$  = thickness of louver board, inches.
- $L'$  = width of louver board, inches.
- $L$  = length of louver board, inches.

These variables can be organized into six dimensionless groups. One of these groups namely,  $dL/2 (L - d)$ , can be neglected for small values of  $d$ , since, when  $d$  is small, this quantity takes on the value,  $d/2$ . The remaining dimensionless groups are:

- $d\rho v/\mu$  = pseudo Reynolds number\* (See footnote page 531).
- $P/\rho v^2$  = resistance coefficient.
- $L'/s$  = width coefficient.
- $d/(d + t)$  = free area ratio.
- $d/s$  = angle coefficient.

For the condition of zero overlap it may be shown that one of the three dimensions  $d$ ,  $s$ , or  $L'$  is not an independent variable because given any 2 of the 3 variables the third is fixed. One of these variables may thus be eliminated in considering this type of louver. The choice made was to eliminate the angle coefficient  $d/s$ , thus giving 4 dimensionless groups for this type of louver.

#### CALCULATIONS

The following formula was used to calculate air velocity from Pitot tube readings.

$$v' = 1096.5 \sqrt{\frac{h_v}{\rho}}$$

in which

$v'$  = velocity of air at point of measurement, feet per minute.

$h_v$  = velocity pressure, inches of water.

$\rho$  = density of air, pounds mass per cubic foot.

Essentially the same formula was used for the nozzle except that it was necessary to include a nozzle discharge coefficient and a correction factor for approach velocity.

The nozzle discharge coefficient was obtained from an alignment chart made by the Texas Engineering Experiment Station, and based on *ASME Transactions*.

The approach velocity correction factor was obtained from the relation

$$C = \frac{1}{\sqrt{1 - \left(\frac{A_n}{A_D}\right)^2}}$$

in which  $A_n$  is the throat area of the nozzle, and  $A_D$  is the area of the duct.

The volume of flow was determined by the expression  $Q = Av'$ , in which  $A$  equals  $A_n$  for the nozzle and  $A_D$  for the Pitot tube.

The average velocity on the inlet side of the louver was then obtained from  $v = Q/A_e$ , in which  $A_e$  is the cross-sectional area of the entrance duct.

The density of the air was obtained from an Air Density Calculator designed by the Texas Engineering Experiment Station. This calculator is a slide rule device which gives air densities for different dry-bulb and wet-bulb temperatures and barometric pressure.

The viscosity of the air was obtained from a graph plotted from data given in published tables.

The calculation of the dimensionless groups,  $dv\rho/\mu$  and  $P/\rho v^2$ , required the use of several conversion factors because of the inconsistent units of the variables,  $P$ ,  $v$ ,  $d$ ,  $\rho$ , and  $\mu$ . The group  $dv\rho/\mu$  has the units,  $\text{ft}^2\text{-in.-lb}_m\text{-sec}/\text{min-ft}^3\text{-lb}_m$ . To make this group unitless, the necessary conversion factor was obtained as follows:

$$\begin{aligned} dv\rho/\mu &= (\text{ft-in.-lb}_m\text{-ft-sec}/\text{min-ft}^3\text{-lb}_m) (\text{min}/60 \text{ sec}) (\text{ft}/12 \text{ in.}) \\ &= (\text{calculated units}) (1/720) = (\text{calculated units}) (0.0013889). \end{aligned}$$

\* The term pseudo Reynolds number is used here because this group is not a ratio of inertia forces to viscous forces, but rather a function of this ratio. The approach velocity was used because it is common to all of the louvers, is easy to measure, and is related to the velocity within the louver by the continuity equation. If the velocity within the louver were used in place of the approach velocity, it would be necessary to state at what position in the louver this velocity corresponds because this velocity varies (except in the standard louver) throughout the louver. Another alternative is to use the velocity at the position in the louver where  $d$  is measured. Using this velocity,  $v'$ , in the  $dv\rho/\mu$  term results merely in a product of two dimensionless groups, namely  $(dv\rho/\mu) (v'/v) = (dv\rho/\mu) (A/A') = (\text{pseudo Reynolds number}) (1/\text{Free Area Ratio}) = dv\rho/\mu$  so that no new variables have been introduced. It is true that using the  $dv\rho/\mu$  term tends to cause the curves to fall on top of each other, but not to such an extent that a single curve could be used without appreciable error.

To make the group  $P/\rho v^2$  unitless, the necessary conversion factor was found in a similar manner as shown below.

$$P/\rho v^2 = (\text{in. of water} \cdot \text{ft}^3 \cdot \text{min}^2 / \text{lb}_m \cdot \text{ft}^2) (g \text{ ft/sec}^2) \\ (62.3 \text{ lb}_m/\text{ft}^3) (\text{ft}/12 \text{ in.}) (3600 \text{ sec}^2/\text{min}^2)$$

in which  $g$  is the local acceleration of gravity or approximately  $32.11 \text{ ft/sec}^2$  in the College Station, Texas area.

Hence

$$P/\rho v^2 = (\text{calculated units}) [(32.11) (62.3) (3600)/12] = (\text{calculated units}) (600,140).$$

## DISCUSSION

R. W. MCKINLEY, Pittsburgh, Penna., (WRITTEN): What meaning has this study for the architects and engineers who are concerned with the natural ventilation of buildings via louvered fenestration areas in warm weather?

L. T. MART, Kansas City, Mo., (WRITTEN): This paper provides valuable additional information to that contained in Cobb's December, 1953 published paper on "Pressure Loss of Air Flowing Through 45-Degree Wooden Louvers". Louvers used for air intakes on cooling towers may be responsible for as much as  $\frac{1}{8}$  to  $\frac{1}{2}$  of the total pressure loss of some induced draft tower designs, whereas loss through spray eliminators seldom exceeds 10 to 20 percent of the total. (Author mentions 20 percent for each.)

The lower section and the duct to which it is fastened are of the same size, as shown in Figs. 1 and 2. This set-up provides comparative performance data on the 4 types of test louvers, but does not simulate actual operating conditions on a cooling tower, where the air entry is free-flowing from all directions and the flow resistance is considerably less. It would be interesting to know the velocity of air through the louvers for the 4 curves shown in Fig. 5.

The author's results indicate that the inverted type louver should be desirable for installations where pressure loss is a factor. Also that a decrease in the louver spacing indicates a decrease in the width required for low pressure loss. Present cooling tower louver design reflects these opinions; however, one would probably obtain equivalent performance with wider boards and opening up the louver spacing, thus eliminating the expensive bevelling of the louver ends. The correlated economics of design and manufacture are important factors.

This problem of louver design is one of definite interest to cooling tower manufacturers, hence we are glad to see the publication of information on this subject.



**1553**

## PERFORMANCE AND EVALUATION OF ROOM AIR DISTRIBUTION SYSTEMS

By ALFRED KOESTEL\* AND G. L. TUVE\*\*, CLEVELAND, OHIO

**This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with Case Institute of Technology, Cleveland, Ohio.**

**T**HE OBJECT of the designer of a room-air distribution system is to provide ventilation of an occupied space and to absorb heating and cooling loads. This result must be accomplished without drafts in the occupied zones. A general method of analysis of air distribution systems for small rooms is offered here with results of tests of several systems under controlled conditions.

In the early development of both heating and cooling by air systems, it was common to use large temperature differences between the supply air and the room air and to pay little attention to air distribution. The adverse comfort conditions of such systems are now recognized. But when low temperature differences and large air volumes are used, the problem of drafts becomes very important.

This paper attempts to establish a method for determining optimum comfort conditions for high-volume convection systems. Special attention is paid to heating with low supply temperatures, as in heat pump applications.

### EVALUATION CRITERIA

Available data on health and comfort are insufficient to give a fully satisfactory basis for evaluating room-air conditions. In this paper the findings of Houghten<sup>1</sup> and his associates and the proposals of Rydberg and Norback<sup>2</sup> were used for the evaluation of effective temperatures and drafts (see Terminology for definitions).

One criterion of good air distribution is a uniform feeling of warmth throughout the occupied zone, *i.e.*, uniform effective temperature due to dry bulb and air velocity. A physiological sensation (change in skin temperature) caused by a change in local effective temperature is therefore called a *draft*. Excessive local air velocities may also be in themselves objectionable, causing movement of the hair, eye discomfort, disturbance of papers or fabrics or directional movement of smoke. In any case it is recognized that objectionability or tolerability are statistical with respect to the occupants. Each given condition will result in a certain percentage of objections, and it is very difficult to produce *any* condition that is voted satisfactory by all occupants.

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\*\* Professor of Mechanical Engineering, Case Institute of Technology. Member of ASHAE.

<sup>1</sup> Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, San Francisco, June 1955.

Dry-bulb temperature and air motion are the only conditions taken into account in the present evaluation. Humidity is disregarded because extremes of humidity are uncommon in properly conditioned rooms. The mean radiant temperature of all enclosing surfaces is assumed to be within a few degrees of the air temperature in the occupied zone. This is the actual case in most well-constructed buildings served by air distribution systems. Radiant heating and cooling are considered to be methods of absorbing a portion of the thermal load and of compensating for extremes of room-surface temperature due to weather and solar effects.

#### THE IDEALIZED SYSTEM

An actual system of air distribution is too complex for direct analysis, but by means of simplifying assumptions an ideal or theoretical model may be attained. Such a model will reveal the significant variables and thus act as a guide for the experimental attack and presentation. Consider a system in which the following assumptions apply.

1. An ordinary convection system serves an independent room in a well-insulated structure
2. The mean radiant temperature of room surfaces approximates the air temperature, *i.e.*, radiant effects are small compared with convection.
3. Normal indoor relative humidities are encountered, without extremes.
4. At each test condition the system is in equilibrium, and maximum comfort is attained at the control point in the middle of the room.
5. Inertia forces predominate as compared with buoyant forces, *i.e.*, the circulation in the occupied zone is mainly by forced convection rather than by natural convection.
6. In the unoccupied zone, adjacent to exposed surfaces, natural convection is important.
7. Differences between heating and cooling will exist.

Based on these assumptions and on the available comfort data, the following theory has been evolved to describe the performance of a room-air distribution system with forced convection: (See Appendix under Analysis.)

1. At any given point in the occupied zone, the magnitude of the air velocity is a function of the flow rate at the supply openings.
2. The vector direction of the air velocities in a room, *i.e.* the flow *pattern*, is a function of the room configuration and of the type and location of distribution system. There may be a difference between heating and cooling.
3. The percentage of occupants who will object to a local *draft* depends on the departure of the local effective temperature from that at the control point, and this difference  $\theta$  is in turn expressed by the equation:

$$\theta = \Delta t - 0.07V, \text{ (see Terminology and Equation A-4 in Appendix)}$$

where  $\Delta t$  is the departure of the local dry-bulb from control temperature and  $V$  is the local air velocity, fpm. Still air is assumed at the control point.

4. The local dry-bulb temperature difference ( $\Delta t$ ) at any point in the air flow pattern depends on the supply-air temperature difference only, and is independent of normal changes in the air flow rate.

5. For a given room, served by a given system, there is one air flow rate and corresponding supply temperature that will result in minimum draft.

#### TEST ROOM AND EXPERIMENTAL PROCEDURE

The test room was 18 x 13 x 8 ft, built to simulate a well-insulated living room with 2 exposed walls and windows, insulated unheated attic, and an insulated floor

with ventilated crawl space. Conventional furniture was not present. The 18-ft exposed wall included a 10 x 6 ft simulated double-glass picture window and two 5 x 4 ft simulated double-glass windows were located on the 13-ft exposed wall. An outdoor space was provided on the 2 exposed walls as well as above and below the room. Air pressure differences between the test room and outdoor space, and test room and laboratory were reduced to a minimum by damper control. This eliminated any extraneous air infiltrations or exfiltrations. Infiltration air from the outdoor space was metered into the test room from perforated panels located on each side of the picture window. Air was supplied to the air distribution outlets from a separate plenum chamber.

Typical heat losses were obtained by designing the exposed surface of the test room for the same interior surface temperatures encountered in a typical well-insulated house. When an outdoor space temperature simulating 8 F outdoors was maintained, the following typical test room conditions were noted.

1. Room Control Temperature (center of room at the 30-in. level): 73 F.
2. Infiltration Heat Loss: 500 to 650 Btu/hr.
3. Total Register Heat Input: 5000 to 7000 Btu/hr depending on air change and type of air distribution system.
4. Surface Temperatures (see Fig. 1).
5. Total Heat Loss (fixed internal heat gain from lights and occupants of 1,000 Btu/hr plus register heat input): 6000 to 8000 Btu/hr.

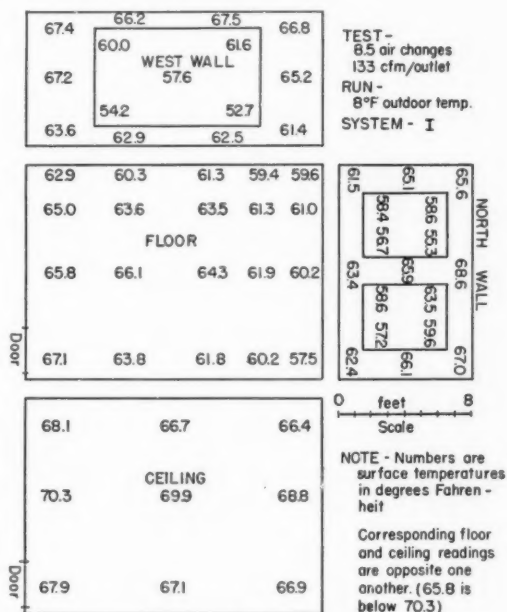


FIG. 1. TYPICAL SURFACE TEMPERATURES MEASURED WITH SYSTEM I, 8 F OUTDOOR TEMPERATURE

TABLE 1—AIR DISTRIBUTION SYSTEMS IN AN 18- x 13- x 8-FT TEST ROOM

Three 8- x 18-in. returns individually tested with all systems of air distribution. One return was located low on the inside wall opposite the picture window, another low on the exposed wall below the picture window, and the third high on the inside wall near the corner formed by the two inside walls. When testing a given system, all air distribution outlets were balanced to deliver equal amounts of warm air.

DESIGNATION IN FIGURES	AIR DISTRIBUTION SYSTEM	NO. OF OUTLETS	SIZE OF OUTLET	LOCATION OF AIR DISTRIBUTION OUTLETS
I	High Side Wall Outlets	2	12 in. x 6 in. approximately 48 sq. in. free area per outlet. The vertical bars were set to produce a uniform total spread of 60 deg in the outflowing air.	High on the inside wall opposite the picture window with their centers one foot from the ceiling and one quarter of the length of the room from the adjacent walls.
II	Baseboard Diffusers	3	30 inches in length 4½ in. in height approximately 26.5 sq in. free area	Two of the three baseboard diffusers were located adjacent to the floor on the exposed wall containing the picture window and one quarter of the length of the room from the adjacent walls. The third baseboard diffuser was located adjacent to the floor and centered on the exposed wall containing two 4- x 5-ft. windows.
III	Ceiling Diffusers (Flush Cone Type)	2	7-in. neck diameter	Centrally located each in a 9- x 13-ft rectangle which constituted one-half of the ceiling area.
IV	Ceiling Diffusers (Projecting Cone Type)	2	6-in. neck diameter	Centrally located each in a 9- x 13-ft rectangle which constituted one-half of the ceiling area.
	Perforated Acoustical Baffles as Air Diffu- sion Outlets*	2 to 5	4 ft long by 6 in. in height approximately 1 sq ft of free area and 4 sq ft of gross surface area	A complete baffle system was installed. The five baffles nearest to the cold walls were fitted with air supply ducts and the number of baffles tested were varied from 2 to 5.

\* NOTE: Wedge-shaped perforated aluminum trough filled with a glass fiber core for sound absorption used in connection with luminous ceiling.

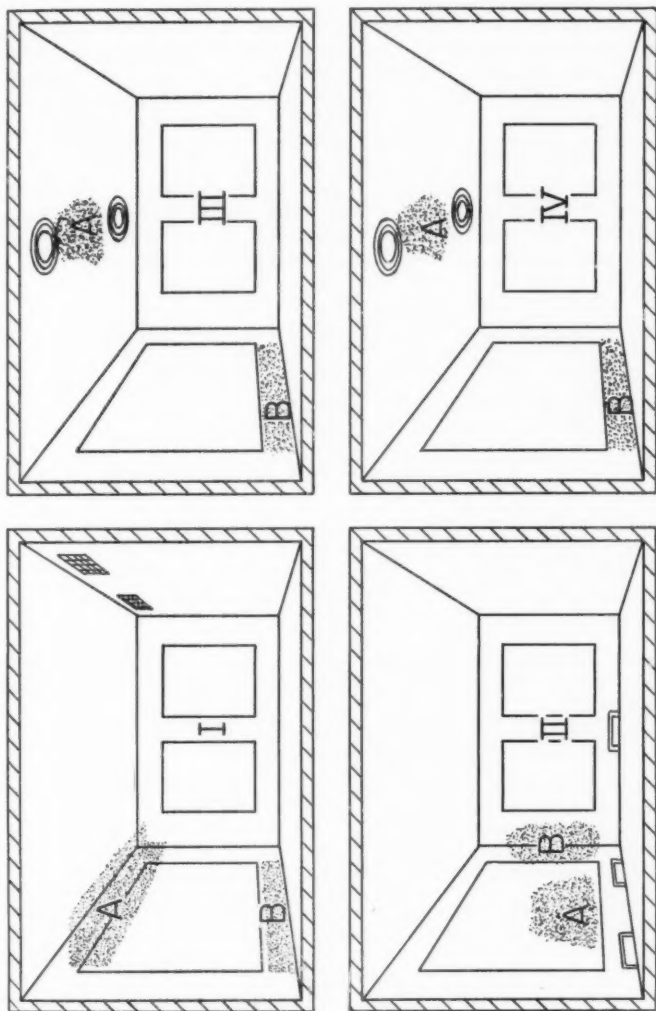


FIG. 2. TEST ROOM SHOWING WINDOWS, SUPPLY OUTLETS AND CRITICAL DRAFT REGIONS A AND B (SEE TABLE 1). ROMAN NUMERALS DESIGNATE AIR DISTRIBUTION SYSTEMS

The occupied zone of the test room was defined as being enclosed by surfaces 1 ft from the walls and 6 ft, 2 in. from the floor. The boundaries of the occupied zone were marked by a grid consisting of strings arranged in 6-in. squares with calibrated air-movement indicating threads in each square.

Table 1 and Fig. 2 provide a description of the 5 air distribution systems tested.

For each system of air distribution and for each constant outdoor temperature, air at various temperatures was introduced into the test room. The quantity of air in each case was that required to maintain the room control temperature at 73 F. The supply duct dampers were adjusted so that for each system of air distribution the outlets were balanced and delivered equal quantities of air. This was repeated for different outdoor temperatures.

After equilibrium had been obtained for each set of conditions, air distribution measurements were made on the grid. Since only the regions on the grid having critical draft conditions were considered important, these regions were located and probed with calibrated anemometers and temperature measuring instruments.

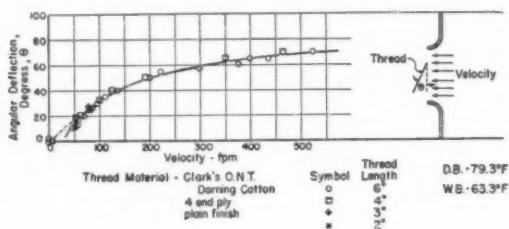


FIG. 3. CALIBRATION OF VELOCITY-INDICATING THREADS. IDENTICAL THREADS LOCATED IN EACH 6-IN. SQUARE BOUNDING OCCUPIED ZONE. CALIBRATED WITH AIR AT 79.3 F DB AND 63.3 F WB

This technique limited draft measurements in the room air flow pattern to the boundaries of the occupied zone where the draft would always be found to be the most objectionable.

In conjunction with the air temperature and air velocity measurements in the critical draft regions, the following data were obtained.

1. Surface temperatures on exposed walls, windows, ceiling and floor.
2. Register and return air temperatures and air flow rates.
3. Outdoor space temperatures.
4. Quantity and temperature of infiltration air.
5. Vertical air temperature gradients throughout the occupied zone.

Three returns were separately tested in order to determine the effect of return location on the air and temperature distribution.

#### INSTRUMENTS

Anemometer measurements were made by several methods: (1) from angular deflections of calibrated, air movement-indicating threads (see Fig. 3); (2) by commercial heated-thermocouple anemometers; (3) by heated thermometer anemometer; (4) by heated thermocouple anemometer as developed by Nottage at

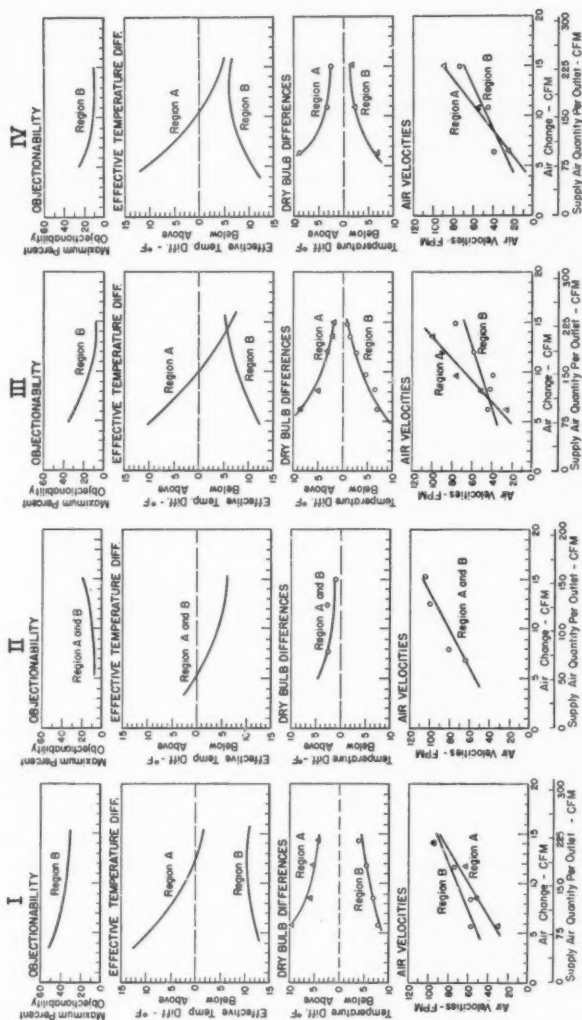


FIG. 4. AIR AND COMFORT CONDITIONS IN CRITICAL DRAFT REGIONS A AND B, SYSTEMS I TO IV AT 8F OUTDOORS, 73F CONTROL.

the ASHAE Laboratory. All anemometers were calibrated in the minimum diameter section of a bellmouth intake nozzle. In addition, the calibration was checked on the discharge end of a converging nozzle mounted on a plenum.

Provisions were made to measure the floor-to-ceiling air-temperature gradients at various positions in the occupied zone by means of thermocouples on vertical poles. Both thermocouples and thermometers were used as temperature probes in the critical draft regions. Surface temperatures on the cold walls, windows, floor, and ceiling were measured by thermocouples attached flush to the surface.

### TEST RESULTS

The results obtained from the tests for room heating are presented graphically to illustrate the general behavior of an air distribution system as brought out by the equations (see Appendix). These results apply when low returns are used except that high returns may also be used in system II.

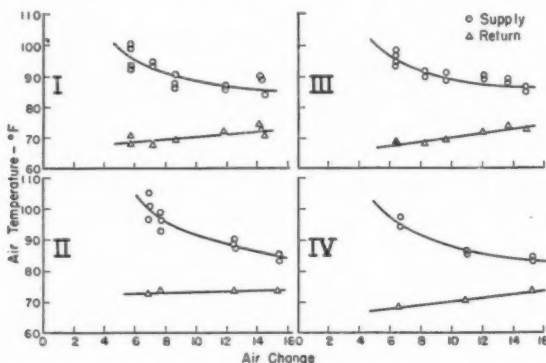


FIG. 5. SUPPLY AND RETURN TEMPERATURE AS A FUNCTION OF AIR FLOW RATE, FOR SYSTEMS I TO IV. 8 F OUTDOORS, 73 F CONTROL

The air velocities in the critical draft regions of the occupied zone plotted as a function of air flow rate or air change are shown in Fig. 4. Note the linear relationship between air change and air velocity in accordance with Equation A-1. Equation A-1 applies to a fixed point in the flow pattern whereas the experimental data apply to a region on the boundaries of the occupied zone covering an area of, say, 10 sq ft.

For a given room heat load, as the room-air velocities increase with air change, the air temperature differences in the critical draft regions with reference to the control temperature decrease according to Equation A-2 and as shown in Fig. 4.

The degree of draft in the critical draft region with respect to the control point and as measured by the effective temperature difference is plotted in Fig. 4. If the air temperature in the critical draft region is above the control point temperature, a *no draft* ( $\theta = 0$ ) condition can be obtained. If the air temperature is below the control point temperature, a minimum draft condition (i.e. minimum  $\theta$ ) will result as shown in Fig. 4 and according to Equation A-8.

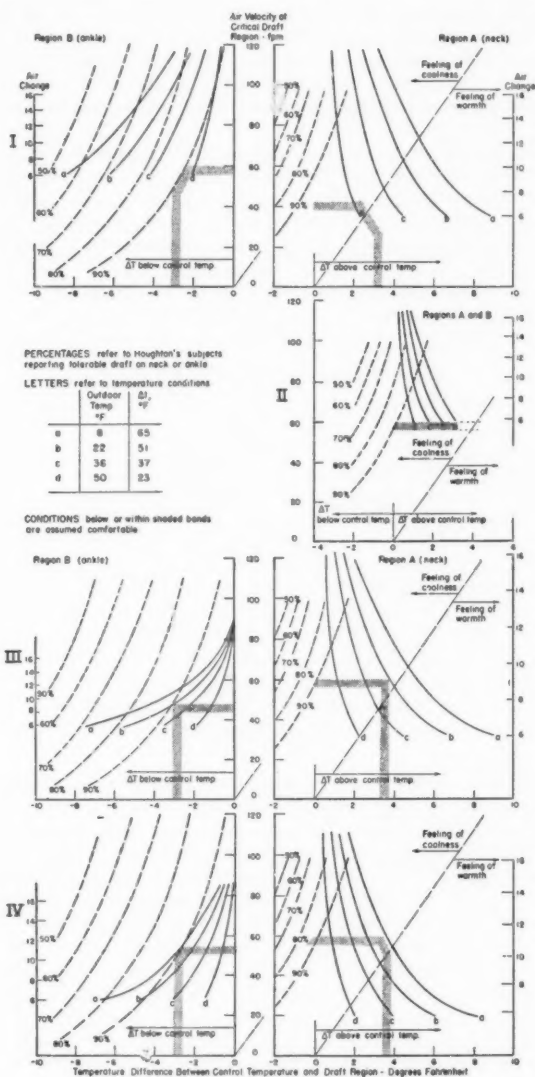


FIG. 6. DRAFT CHARACTERISTICS OF SYSTEMS I TO IV, VARIOUS FLOW RATES AND OUTDOOR TEMPERATURES

The probable percentage of occupants who would object to the draft in the critical draft region in accordance with Houghten's findings (see Terminology) is shown in Fig. 4. Note that the general trend of the objectionability curve is similar to the curve of effective temperature difference, as should be expected.

In Fig. 5 are plotted the supply-air temperature and return air temperature as a function of air change.

In Fig. 6 are curves plotted in accordance with Equation A-2 in order to show the general behavior of draft in the critical draft regions. The air velocity is plotted against the air temperature difference for various indoor-outdoor temperature differences which is in effect for various values of  $Q_0$  in Equation A-2. The curves apply to the critical draft regions, and superimposed on these are the curves of equal tolerability, so as to illustrate the probable reaction of occupants to the draft as the outdoor temperatures and air changes are varied for various systems of air distribution.

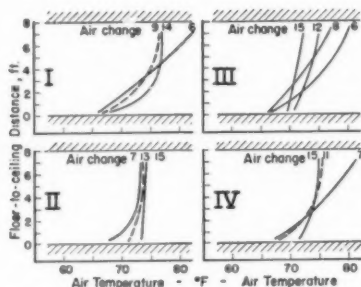


FIG. 7. TYPICAL VERTICAL AIR TEMPERATURE GRADIENTS NEAR THE PICTURE WINDOW, 8 F OUTDOORS, VARIOUS RATES OF AIR FLOW

Typical vertical air temperature gradients in the region near the picture window are shown in Fig. 7. In Fig. 8 typical temperature differences between head and ankle levels are plotted versus the temperature differences between supply and return air. The curves are given for the air-distribution systems I to IV in accordance with Equation A-3. These curves apply to the region near the picture window as well as for any heating load and air change.

When heating with perforated baffles, the number of baffles in use and their location (see Table 1) may influence the general air flow pattern in the space and as a result one would expect the constant  $C$  in Equation A-3 to depend on the number of baffles,  $N$ . In addition the relatively large size of a baffle and consequent low air velocity may magnify the buoyant effects and this complicates the simple relationship defined by Equation A-3. The effect of the number of baffles and buoyancy on the air temperature difference can be determined experimentally by evaluating the following general function which is just an extension of Equation A-3:

$$\Delta t / \Delta t_o = f \left[ \frac{(M_o / N)^2}{\Delta t_o}, N \right] \dots \dots \dots (1)$$

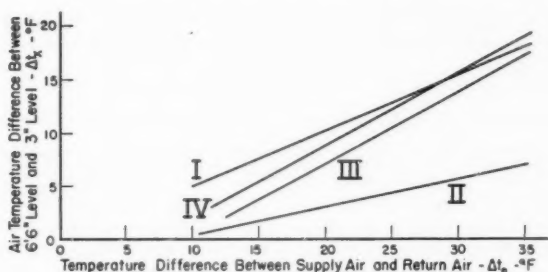


FIG. 8. TEMPERATURE DIFFERENCES 3 TO 78 IN. FROM FLOOR, NEAR PICTURE WINDOW, ANY FLOW RATE OR OUT-DOOR TEMPERATURE, SYSTEMS I TO IV (SEE EQUATION A-3)

where the ratio  $(M_o/N)^2/\Delta t_o$  is proportional to the ratio of the momentum forces to the buoyant forces, evaluated at the outlet of the perforated baffles. In Fig. 9 the vertical air temperature difference measured near the picture window is plotted in accordance with Equation 1. Note that for a given number of baffles the ratio  $\Delta t/\Delta t_o$  depends on  $(M_o/N)^2/\Delta t_o$  only in the low range, (low momentum force and large buoyant force). For large values of  $(M_o/N)^2/\Delta t_o$ , the ratio  $\Delta t/\Delta t_o$  appears to depend only on the number of baffles.

Preliminary tests on room cooling with perforated baffles showed room air temperature gradients which were appreciably smaller than those obtained for room heating with the same air supply temperature differences. This is primarily due to the fact that with a cooling load the warm air rising on the surface of the window mixes with the cold supply air before entering the occupied zone. With a heating load the cold air spilling down from the windows mixes with the supply air stream near the floor resulting in larger air temperature gradients. Baseboard diffusers will, of course, counteract this phenomenon.

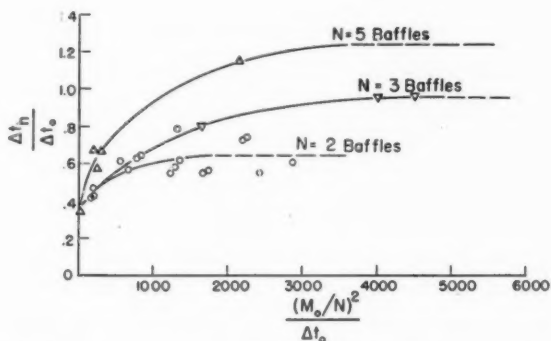


FIG. 9. EXPERIMENTAL RESULTS WHEN USING ACOUSTICAL BAFFLES, (SEE EQUATION 1)

The maximum permissible air quantity per baffle was found to be 130 cfm which produced an air velocity of 60 fpm directly under the baffles at the 6-ft, 2-in. level of the occupied zone.

#### EFFECT OF RETURN LOCATION ON AIR AND TEMPERATURE DISTRIBUTION

Each air-distribution system was tested with 3 different locations of the single return (see Table 1). For each system the return location was changed and the outdoor temperature maintained constant. The heat input was adjusted to maintain the same room control temperature for each run. (1) For the baseboard diffuser system there was no significant difference in gradient, return temperature, register temperature or heat input for any of these locations of the return. (2) With the high sidewall outlets at 12 air changes there was little difference between the 2 low returns but with the high return, the register temperature, return temperature and vertical air temperature gradients were all considerably higher. (3) With the acoustical baffles appreciable increases in vertical air temperature gradients were noted with the high return. (4) With the ceiling diffusers at 15 air changes there were no significant differences in vertical air temperature or air velocities between the 3 return locations. At a lower air change of 6 the 2 low-sidewall returns gave similar performances, but with the high-sidewall return the supply temperature, return temperature and vertical air temperature gradients were all considerably higher.

In all tests, varying the return location did not affect the critical draft velocities appreciably. The effects caused by varying the return location can be attributed to *short-circuiting* of the warm air supply. If the distribution system produces small temperature gradients with floor returns the temperature of the air entering the return is relatively independent of the return location. If, however, the gradients are large with floor returns, mixing is incomplete and changing to a ceiling return will result in withdrawing air at a temperature above the normal room temperature. Since mixing is poor, (evidenced by the high gradient with floor returns), the withdrawing of the warm air near the ceiling will aggravate the situation, resulting in increased gradients, return temperature, register temperature and heat input.

#### EVALUATION AND CONCLUSIONS

The following conclusions apply to the type of installation under consideration *i.e.*, a high volume system for heating a typical living room in a well-insulated residence, and to the air distribution systems tested.

The theory predicted that the air velocities in the occupied zone would be proportional to the velocity at the supply outlets, and this was verified by all tests except those using perforated baffles for air supply. In other words, for the common types of registers, diffusers and baseboard slots used in residential heating, the *pattern* of air distribution is fixed by the geometry of the room and system, and the air *velocities* change with the volume of air circulated. For a given supply temperature the room air velocities vary with the indoor-outdoor temperature difference.

Combining these results with the existing information on the reaction of people to effective temperatures and drafts, it is possible to map the performance of four types of air distribution systems for heating, over a wide range of outdoor temperatures and air supply rates, Fig. 6.

In Fig. 6 is also shown a shaded boundary outside of which the performance of each air distribution system is considered unsatisfactory and inside of which the performance is satisfactory.

The values used to define this boundary are based on the following criteria of evaluation:

1. Maximum allowable air movement in the occupied zone of 60 fpm.
2. 90 percent minimum tolerability or 10 percent maximum objectionability to the draft in the occupied zone.
3. Less than 7 deg vertical air temperature difference in the occupied zone between the 6-ft., 6-in. level and the 3-in. level. (Not over 3 deg temperature-difference between draft region B and control).

Actual air velocities and air temperatures produced by a system of air distribution were found by test measurements to approximate the general behavior described by the equations:

$$\begin{aligned} V &= C_1 M_o \text{ (see Fig. 4)} \\ Q_o &= C_2 V \Delta t \text{ (see Fig. 6)} \\ \Delta t_o &= C \Delta t \text{ (see Figs. 8 and 9 and Equation 1)} \end{aligned}$$

A system of air distribution can be evaluated on the basis of three factors: namely, vertical air temperature differences, air velocity, and the objectionability or tolerability of draft in the occupied zone (Fig. 6).

#### ACKNOWLEDGMENTS

The authors gratefully acknowledge the generous financial support of this project by the General Electric Co. Weathertron Department, the F. W. Wakefield Brass Co., the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, and Case Institute of Technology. We are indebted to Prof. L. G. Seigel and Harold Reed for supplying the performance data on the perforated acoustical baffles. The figures were prepared by Mrs. Ruth E. Strahosky.

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2. Air Distribution and Draft, by John Rydberg and Per Norback (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 225).
3. ASHVE RESEARCH REPORT No. 1512—Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air, by Alfred Koestel (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 385).

#### TERMINOLOGY

The following list explains the technical terminology pertaining to this paper:

1. *Occupied Zone*: The region in the test room which is enclosed by imaginary planes 1 foot from all walls and 6 ft, 2 in. from the floor is the occupied zone. In the test room the boundaries of the occupied zone were defined by a grid consisting of a string mesh made up into 6-in. squares. All draft measurements were limited to the plane of the grid, since the most objectionable draft conditions in the occupied zone would occur on the grid surfaces.

2. *Effective Temperature Difference:* The effective temperature difference is a quantitative measure of the difference in physiological feeling of warmth or coolness between 2 points within the test room. Normally, 1 of the 2 points is taken at a fixed reference point at the center of the room at the 30-in. level (referred to as the control temperature). An approximate equation defining the effective temperature difference, neglecting radiation and humidity, is:

$$\theta = \Delta t - 0.07V \text{ (Reference 2)}$$

where

$\theta$  = effective temperature difference between still air at control temperature and draft region.

$\Delta t$  = dry-bulb temperature in draft region minus control temperature.

$V$  = air velocity in draft region, feet per minute.

3. *Room Control Point:* The room control temperature is the dry-bulb temperature in the center of the test room at the 30-in. level. This temperature is measured by a shielded dry-bulb thermometer. This temperature was maintained as close to 73 F as was experimentally possible during all test runs.

4. *Draft:* The term *draft* in this report means any local sense of cooling or warmth of a portion of the body due to air movement and air temperature, humidity having been held constant. A measure of draft is the effective temperature difference. The warmth or coolness of a draft is usually measured in these tests above or below the room control conditions.

5. *Percentage Tolerability:* In predicting probable reaction of occupants to any of these air distribution systems, Houghten's data on the tolerability of drafts is taken as the criteria (Reference 1). The percentage of his subjects who would not object if their necks or ankles were exposed to the draft condition measured in the test room is referred to as the *percentage tolerability* of that draft. The location of a draft was, of course, the determining factor in deciding whether it was the neck or the ankles which would most likely be affected by it, and in choosing one or the other of Houghten's sets of data accordingly. If the draft occurs within 6 in. of the floor surface, the ankles are taken as the reference region, and if the draft occurs some distance above the floor level, the neck is taken as the reference region.

6. *Percentage Objectionability:* The percentage objectionability of a draft is the opposite of its percentage tolerability (Reference 1). It indicates the percentage of people who *would* object to a measured draft condition, and it can therefore be found by subtracting the percentage tolerability from 100 percent. For example, on measuring a draft in the *neck region*, we might find that it would produce a skin temperature drop of 3.3 deg. We could then see from Houghten's figures that a probable 10 percent of the people experiencing this feeling of coolness on their necks would find it objectionable. Therefore, we would say that the measured draft had a percentage objectionability of 10 percent and a percentage tolerability of 90 percent.

7. *Critical Draft Regions:* The critical draft regions are regions in the occupied zone of the test room which might contain drafts of maximum objectionability with respect to the effective temperature experienced at the room-control temperature (still air at 73 F). The critical draft regions of the occupied zone will always be on the grid defining the boundaries of the occupied zone. In the 5 air distribution systems investigated, it was found that the critical draft regions in each system covered an appreciable grid area of at least 10 sq ft.

8. *Air Change* = cfm  $\times$  60/Room Volume in cubic feet.

## Symbols

- $C$  = constants.  
 $C_1$  = constants.  
 $C_2$  = constants.  
 $M_o$  = total air flow rate from all outlets, cubic feet per minute.  
 $(M_o)_m$  = air flow rate for minimum draft, cubic feet per minute.  
 $N$  = number of perforated, acoustical baffles.  
 $Q_o$  = register heat input, Btu per hour.  
 $V$  = air velocity, feet per minute.  
 $\Delta t$  = temperature difference, Fahrenheit.  
 $\Delta t_h$  = vertical air temperature difference between the 6-ft, 6-in. level and the 3-in. level.  
 $\Delta t_o$  = air temperature difference, supply to return, Fahrenheit.  
 $\theta$  = effective temperature difference, Fahrenheit (see Terminology).

## APPENDIX

## THEORY AND ANALYSIS

A given system of air distribution will set up an air flow throughout the space, the pattern of which is, in general, almost independent of the air flow rate at the supply outlet. The magnitudes of the air velocities at the various points in the space can be considered a function of the supply air flow rate, and the vector direction of the air velocities (air flow pattern) at these various points can be considered a function of the type of air distribution system and the space configuration.

At any fixed point in an idealized free air jet, and for given outlet dimensions, jet theory predicts that the air velocity varies linearly with the outlet air flow rate if buoyant forces are neglected or,

$$V = C_1 M_o \text{ (see Reference 3)} \quad \text{A-1}$$

Behavior of actual room air velocities can be expected to approximate Equation A-1 if the air distribution system and space configuration are fixed and inertia forces predominate. When heated or chilled air is introduced into the space, heat is transferred by convection and by mixing or entrainment from the air jet to the enclosing surfaces of the space. It may intuitively be expected that the heat flux in Btu per minute per unit area at given points throughout the air flow pattern should be proportional to the heat flow at the register outlet. This may be expressed simply as,

$$Q_o = 1.06 M_o \Delta t_o = C_2 V \Delta t \quad \text{A-2}$$

where the temperature difference  $\Delta t$  is arbitrarily measured with respect to some datum point, say the control temperature.

Combining Equations A-1 and A-2 and eliminating the jet velocity  $V$ , we have,

$$1.06(M_o \Delta t_o) = C_2(C_1 M_o) \Delta t$$

or

$$\Delta t_o = C \Delta t \quad \text{A-3}$$

Equation A-3 states that temperature differences in the air flow pattern depend only on the supply air temperature difference and are independent of air flow rate.

The effective temperature difference,  $\theta$ , with respect to an arbitrarily selected control point at any fixed point in the air flow pattern can be considered:

$$\theta = \Delta t - 0.07 V \text{ (Reference 2, see Terminology)} \quad \text{A-4}$$

A measure of the tolerability or objectionability of draft can be established by assuming the percentages of Houghte's subjects<sup>1</sup> reporting tolerable or objectionable

draft on the neck or ankle when subjected to various air temperatures and air velocities (see Terminology).

The temperature difference and air velocity Equations A-1 and A-3, can be substituted into the effective temperature difference Equation A-4 in order to obtain,

$$\theta = \Delta t_o / C - 0.07 C_1 M_o \quad \text{A-5}$$

The room heating or cooling load can be expressed by,

$$Q_o = 1.06 M_o \Delta t_o \text{ Btu per hr} \quad \text{A-6}$$

combining Equations A-5 and A-6 we have,

$$\theta = (Q_o / 1.06 C M_o) - 0.07 C_1 M_o \quad \text{A-7}$$

Equation A-7 brings out the fact that there may be a particular air flow rate which will result in minimum draft ( $d\theta/dM_o = 0$ ) or in *no draft* ( $\theta = 0$ ) as the case may be.

If the constants in Equation A-7 were determined from experiment, the air flow rate for minimum draft at a given point in the air flow pattern can be found by setting  $d(\theta)/d(M_o) = 0$  as follows:

$$d\theta = -\frac{Q_o dM_o}{1.06 C M_o^2} - 0.07 C_1 d(M_o)$$

and

$$d\theta/dM_o = 0 = (Q/1.06 C M_o^2) - 0.07 C_1$$

$$(M_o)_m = \sqrt{\frac{-Q_o}{1.06 \times 0.07 C C_1}} \quad \text{A-8}$$

Equation A-8 can be solved for the air flow rate,  $(M_o)_m$ , resulting in minimum room draft if the heating or cooling load is known, and if the constants  $C$  and  $C_1$  for the region containing the most objectionable draft can be evaluated. Note that minimum draft values are obtained from Equation A-8 only if the constant  $C$  is negative, which means that  $\Delta t$  in Equation A-3 is the difference in air temperature measured below the control temperature.

## DISCUSSION

H. E. STRAUB, Urbana, Ill., (WRITTEN): This paper is of special interest because of the suggested indices for evaluating room air distribution. In the final analysis, the purpose of any study on air distribution is to predict possible comfort reactions from such measurable variables as velocity, temperatures and humidity. Future physiological research may show new correlating values between comfort and the measurable variables, but the indices show a method of using the present knowledge to estimate the probable comfort reactions. Especially during cooling such indices are needed to evaluate the performance because vertical temperature differences alone are an unsatisfactory index.

In an actual installation the occupant will attempt to maintain a zero objectionability at some given point, and different occupants will maintain different control conditions. However, it is reasonable in simulated tests such as these to maintain a constant control temperature in the center of the room, and to evaluate the distribution by an index which compares the differences in conditions in various parts of the room to the conditions at the control point. It is then assumed that these same differences would occur even though one occupant might maintain a slightly higher or lower control temperature than another occupant.

It seems that for the foregoing assumption to be true all of the variables included in an index must be evaluated at the control point as well as at the draft region. In the indices of effective temperature difference and objectionability, the authors have not taken into account the velocity at the control point. The following examples show the effect of the control velocity on the indices:

1. Assume the velocity at the control point to be 30 fpm and at the draft region to be 60 fpm and the draft region temperature to be 78 F.

By the authors' method the effective temperature difference between the two points would be 1 deg and by accounting for the velocity at the control point the difference would be 3 deg. This is done by using another effective temperature as a reference or replaces  $V$  in the authors' equation with the difference in velocity between the draft region and the control point.

2. Assume the same conditions as above except that the draft region temperature is also 73 F.

From Fig. 6 for the neck, the tolerability at the control point is about 95 percent and at the draft region about 88 percent. Because the occupant is satisfied with the control conditions, his objectionability to the draft region would then be the difference between the tolerability at the two points, or 7 percent. The authors would indicate an objectionability of 12 percent for the draft region.

Although, in general, accounting for the velocity at the control point would result in larger effective temperature differences and less objectionability than given, the trends shown by the authors would not be changed substantially. This change is recommended for the authors' consideration as a possible closer link between the measurable variables and comfort.

Regarding the test data, I would like to emphasize that these data apply to a system utilizing high flow rates or low supply-air temperatures. For example, in Fig. 4 with arrangements I, III, and IV for draft region B, if the temperature difference and velocity curves were redrawn through the test points, they would become nearly horizontal below 8 air changes per hour. In other words, the linear relationships given in equations A-1 and A-3 are true for region B only with 8 or more air changes per hour.

From the air distribution studies conducted at the University of Illinois it was found that with these types of outlets with low flow rates a large stagnant layer separated the air in the upper and lower portions of the room. Thus at lower flow rates the velocities and temperatures in region B (near the floor and exposed wall) would be largely a result of natural convection currents, and could hardly be expected to be linearly related to the supply-air conditions.

The studies at Illinois have not indicated such a simple relationship between load, flow rate, and temperature differences as implied in Fig. 6 and Fig. 8 of the paper. Furthermore, the studies reported in the paper were conducted with constant area supply outlets and thus the separate effects of flow rate and supply-air velocity have not been considered.

These data emphasize quite clearly that with different types of outlets limitations should be placed on the load or supply temperature and the flow rates from each type.

I compliment the authors on their method of evaluation, and feel that they have presented data which can be very useful in the design of convection systems utilizing high flow rates

**AUTHORS' CLOSURE (Professor Koestel):** The authors are grateful to Mr. Straub for his valuable contribution to the discussion of this paper. His comments are based on extensive experience in room air distribution research, and should form an important supplement to this paper.

The authors cannot disagree with Mr. Straub's opinion that the control-point air velocity should be included in the determination of the effective temperature difference between control point and the draft region. However, for simplicity a quasi-control point was assumed to exist consisting of still air but having the measured air temperature. This assumes that occupants are satisfied with still air at the control tempera-

ture and that they make their physiological draft evaluation against the assumed control point. In other words, one can consider the occupants coming from a room having still air at the control air temperature and going into the test room to make their draft evaluation. Mr. Straub's method of determining the control point effective temperature would only change the way in which the draft evaluation should be pictured. This we believe is arbitrary.

In the tests presented and in tests now being conducted at Case, we have found that buoyancy becomes important in certain areas, especially at the lower flow rates. This accounts for the reason the curves of Fig. 4 have not been extended below five air changes. We have found that velocities of 20 to 30 fpm exist in draft region B (below picture window) when the air change is reduced to zero. These velocities are no doubt caused by buoyancy, and Equations A-1 and A-3 do not apply in this region. The effect of buoyancy on air temperature difference when using the acoustical baffles is illustrated in Fig. 9 by plotting against the buoyancy parameter  $(M_o/N)^2/\Delta t_o$ . The horizontal part of the curves of Fig. 9 indicates the linear relationship between  $\Delta t_h$  and  $\Delta t_o$  as given by Equation A-3.



**1554**

## ASHAE CODE FOR TESTING AND RATING HEAVY DUTY FURNACES AND DIRECT-FIRED UNIT HEATERS\*†

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\* Adopted 1955 by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.

† This Code was prepared by the ASHAE Committee on Code for Testing and Rating Heavy Duty Furnaces: E. K. Campbell, Chairman; Bowen Campbell, R. S. Dill, A. P. Loeb, C. J. Low, W. J. MaGill, B. F. McLouth, F. L. Meyer, L. G. Miller, A. A. Olson, H. A. Pietsch, O. J. Ress, H. A. Soper.

## SECTION I—PURPOSE

This code describes an apparatus and proposed procedure for testing and rating heavy duty warm air furnaces and direct-fired unit heaters.

## SECTION II—SCOPE

The code is applicable to forced warm air furnaces and direct-fired unit heaters having output ratings in excess of 250,000 Btu per hr, and designed for use with various solid, liquid, or gaseous fuels interchangeably or with a single fuel for commercial, industrial, or heavy duty use.

Other requirements are available for central heating appliances, domestic conversion burners, and unit heaters, as defined in American Standard Installation of Gas Piping and Gas Appliances in Buildings, ASA Z 21.30, 1954 (NBFU Pamphlet 54). Such other requirements when applicable should be used instead of this code.

## SECTION III—DEFINITIONS

**3.1 A Forced Warm Air Furnace** is a device whereby air, heated by combustion of a fuel, is delivered to a building through a system of ducts; the essentials of such a furnace are a casing, a heat exchanger, and fuel burning means such as grate bars or stoker for solid fuel, an oil burner or gas burner, and a fan.

**3.2 A Direct-Fired Unit Heater** is a forced warm air furnace that delivers warm air to the space being heated without the use of a duct system. Turning elbows, stub ducts, or deflecting vanes to direct the discharge of the warmed air are typical parts of unit heaters.

**3.3 The Heat Exchanger** consists of a combustion chamber and any auxiliary heating surface enclosed with it in the same casing.

**3.4 The Combustion Chamber** is that enclosure in which fuel or gaseous derivatives of fuel are burned.

**3.5 Radiator or Economizer** is a term often used to designate a metal enclosure within the same casing as the combustion chamber to which it is connected to form auxiliary heating surface.

**3.6 Primary Heating Surface** is that portion of the heat exchanger exposed to direct radiation from the fire. (The heat transfer surfaces of the combustion chamber are often called primary heating surfaces although this definition may not be exact.)

**3.7 Secondary Heating Surface** is that portion of the heat exchanger not exposed to direct radiation from the fire. (The surface of the radiator or other auxiliary heating surface is often called secondary heating surface although this definition may not be exact.)

**3.8 The Firebox Liner** is that metal or ceramic material installed near the bottom of the combustion chamber to shield it from the heat and chemical action due to primary combustion.

**3.9 The Casing** is that structure which encloses the heat exchanger and forms a passage around it for the air being heated. The fan is either connected to the cas-

ing or enclosed within it. In this latter condition the return air passes through the casing to reach the fan.

**3.10 The Plenum** is that part of the casing from which the warm air ducts receive heated air. The upper part or top of the casing is also called the bonnet.

**3.11 A Radiation Shield** is a metal sheet or plate supported between the heat exchanger and the casing to protect the casing from the heat radiating from the heat exchanger.

**3.12 Standard Air** is air with a density of 0.075 lb per cu ft. This is approximately the weight of dry air at 70 F or of air with 50 percent relative humidity at a dry bulb temperature of 68 F when the barometric pressure is 29.92 in. of mercury. The specific heat is assumed to be 0.240 Btu per (lb) (F deg) for purposes of this code.

**3.13 The Air Delivery** is the rate of discharge of standard air in cubic feet per minute (cfm) from the furnace or unit heater casing outlet or outlets.

**3.14 The Heat Input** is the gross heating value of the fuel supplied to the furnace or unit heater expressed in Btu per hour.

**3.15 The Heat Output** is the heat input less the flue gas loss expressed in Btu per hour.

**3.16 The Gross Output** is the highest output obtained under test conditions described in Section V without exceeding one or more of the rating limits stated in Section IV.

**3.17 The Rating** of a furnace or unit heater is an output, equal to or less than the gross output, which may be selected by the manufacturer for publication.

**3.18 The Efficiency** is the ratio of the output to the input expressed in percent.

**3.19 The Flue Gas Loss** is the heat escaping in the flue gases. It includes the sensible heat in the dry gases, the latent heat of the water vapor in the flue gases and the heating value of any unburned combustible in the flue gases.\*

**3.20 Steady State** is defined for purpose of this code as operation of the furnace or unit heater under such conditions that the air temperature rise does not vary by more than 3 deg, and neither the fuel input rate nor the flue gas loss varies by more than 3 percent at any time during the test.

#### SECTION IV—RATING LIMITS

The rating limits which determine the Gross Output are given in paragraphs 4.1 to 4.5.

**4.1 Flue gas temperature** shall not exceed the inlet air temperature by more than 730 deg. It shall not be less than 300 deg above the inlet air temperature.

**4.2 The heat exchanger surface temperature** at any point shall not exceed the inlet air temperature by more than 930 deg if the surface is grey iron casting or low carbon steel. Higher spot temperatures are permitted when evidence is

\* Note: No CO permitted in test. See Section IV, Paragraph 4.4.

presented that the metal will withstand them. Spot temperatures for alloy and stainless steels shall not exceed a temperature which is 100 deg below the scaling temperatures shown in Table 1 unless scaling temperatures from other reliable sources are furnished by the manufacturer.

**4.3 The casing surface temperature** shall not exceed the inlet air temperature by more than 90 deg when the casing temperature is measured as described in Section V, except at points within 6 in. of the flue outlet collar or other openings through the casing.

TABLE 1—PUBLISHED SCALING TEMPERATURE OF VARIOUS TYPES OF ALLOY AND STAINLESS STEELS<sup>a</sup>

TYPE	SCALING TEMPERATURE
302	1600
303	1600
309	2000
310	2000
316	1650
321	1600
347	1600
410	1250
416	1250
430	1500
446	1900

<sup>a</sup> From *Metals Handbook*, 1948 Edition, Table VII, p. 556.

**4.4 Carbon monoxide (CO)** shall not be present in the flue gases in quantities measurable with the three-pipette Orsat apparatus required in Section V.

**4.5 The temperature rise** of the air shall not be less than 50 deg, nor more than 85 deg. A steady state with reference to temperature rise shall prevail throughout the test. Manufacturer shall state the intended or design temperature rise in the test report.

#### SECTION V—TESTING EQUIPMENT AND METHOD

**5.1 The specimen** tested shall be either the identical furnace or unit heater to which the rating is to apply or a representative sample and true duplicate of furnaces or of unit heaters regularly produced by the manufacturer.

**5.2 The heat exchanger surface temperature** shall be measured by means of at least 10 thermocouples attached to it at the hottest areas. The thermocouples shall be about one foot apart and may be attached by welding, brazing or peening.

**5.3 Casing surface temperatures** shall be measured by means of a contact thermocouple. This thermocouple shall be attached to the center of a disc of thermal insulating material not less than  $\frac{3}{8}$ -in. in diameter.

**5.4 Flue gas temperature** shall be measured by means of a thermocouple grid placed in the flue pipe within one foot of the flue outlet collar. The grid shall consist of at least 6 thermocouples in a plane normal to the gas flow. The thermocouples shall be regularly spaced on a circle concentric with the flue pipe and hav-

ing a radius of two-thirds of that of the flue pipe. There shall be no draft regulator, damper, opening, or exhaust fan between the furnace and grid during the test.

**5.5 A three-pipette Orsat apparatus** shall be provided for determining the proportions of carbon dioxide, oxygen, and carbon monoxide in the flue gas. It shall be connected to draw samples from the flue pipe at the plane of the thermocouple grid.

**5.6 A liquid manometer** or a calibrated gage shall be provided for measuring the draft. The draft shall be measured in the flue pipe, near the smoke collar, and also in the combustion chamber.

**5.7 The inlet air temperature** shall be measured with one or more thermocouples located in the air intake, upstream from the fan.

**5.8 The discharge air temperature** shall be measured by means of a thermocouple grid installed in the outlet duct in a plane normal to the air flow with one thermocouple in each 36 sq in. segment of the duct area. More than 40 segments shall not be required, but where 40 segments are needed they shall be rectangles of equal area. There shall be an elbow followed by a duct having a length equivalent to at least four elbow diameters (or heights) long between the furnace and the grid to protect the thermocouples from heat exchanger radiation.

**5.9 Oil fuel** shall be weighed by means of a scale having a precision equal to at least one percent of the fuel burned per hour. The fuel suction line shall be suspended free from the container during the weighing operation.

**5.10 Oil fuel** used for the test shall be of approximately 35 API degrees at 60 F. See National Standard Petroleum Oil Tables, *National Bureau of Standards, Circular 410* and *Publication M-97*. (See extracts in Tables 2 and 3).

**5.11 Gas fuel** shall be measured with a calibrated meter.

**5.12 Fuel pressure for oil burning furnaces** shall be measured ahead of the nozzle with no valve or other restriction between the nozzle and the gage connection.

**5.13 Fuel pressure for gas burning furnaces** shall be measured at the meter and in the burner manifold (or as near the burner orifices as practicable).

**5.14 Fan speed and motor speed** shall be determined with a revolution counter, a tachometer or a stroboscope.

**5.15 The energy input** to the fan motor shall be measured with suitable meters.

## SECTION VI—TESTING PROCEDURE

**6.1 The furnace or unit heater** shall be assembled in accordance with the manufacturer's instructions and fitted with testing equipment as described in Section V.

**6.2 The furnace or unit heater** shall be operated during a preliminary period to determine maximum firing rate and air temperature rise for its maximum output as defined in paragraph 3.16.

**6.3 The furnace or unit heater** shall be operated, following the preliminary period, at steady state conditions for a period of at least two hours at a selected output equal to or less than its maximum output.

**6.4 Data to identify the device** tested shall be entered as items 1 to 6 in **Section A**, Physical Data of the Furnace or Unit Heater Test Report. Fuel data and the manufacturer's rating data shall be entered as items 7 to 10 in **Section B**, Fuel and Rating Data.

**6.5 Test readings** shall be recorded as items 13 to 30.1 in **Section C**, Test Readings.

**6.6 Computations** shall be made in accordance with **Section D**, Computations and Results, and entered as items 31 to 38.

## SECTION VII—RATING PROCEDURE

**7.1 The furnace rating** shall be calculated in accordance with item 37 of **Section D**, Computations and Results.

**7.2 Air delivery** shall be calculated in accordance with item 38 of **Section D**, Computations and Results.

**7.3 The rating** of the furnace or unit heater for burning solid fuel shall not exceed its rating for burning oil fuel. Tests for ratings with solid fuel are not required.

## FURNACE OR UNIT HEATER TEST REPORT

### Section A—Physical Data

1.	Identification.....	
1.1	Manufacturer.....	
1.2	Model No.....	
2.	Physical Dimensions.....	
2.1	Heating Surface, Total.....	sq ft
2.1.1	Combustion Chamber, Net.....	sq ft
2.1.2	Extended Surface Affixed to Combustion Chamber.....	sq ft
2.1.3	Secondary Surface.....	sq ft
3.	Combustion Chamber Volume.....	cu ft
4.	Material of Combustion Chamber.....	
4.1	Type No.....	
4.2	Scaling Temperature (from Table 1).....	F
5.	Fuel Burning Equipment.....	
5.1	Manufacturer.....	
5.2	Type.....	
5.3	Kind of Ignition System.....	
5.4	Fuel Oil Nozzle, Make.....	
5.4.1	Rating.....	gph
5.4.2	Spray Angle.....	deg
5.5	Gas Orifice.....	
5.5.1	Number of Orifices.....	
5.5.2	Size of Orifices.....	
6.	Fan and Motor.....	
6.1	Name Plate Data.....	
6.1.1	Voltage.....	volts
6.1.2	Amperage.....	amps
6.1.3	Horsepower.....	hp
6.1.4	Fan or Blower Speed.....	rpm
6.1.5	Motor Speed.....	rpm

## Section B—Fuel and Rating Data

7.	Fuel Oil used for Test.....	
7.1	Gross Heating Value.....	Btu per gal
7.2	Gross Heating Value.....	Btu per lb
7.3	Viscosity, Saybolt Universal.....	secs at 100 F
7.4	API Degrees.....	at 60 F
7.5	Flash Point.....	F
7.6	Fire Point.....	F
7.7	Ultimate Analysis.....	% by weight
	C.....	
	H.....	
	N.....	
	O.....	
	S.....	
8.	Gas Fuel used for Test (Kind ——).....	
8.1	Gross Heating Value.....	Btu per cu ft
8.2	Specific Gravity.....	
8.3	Ultimate CO <sub>2</sub> .....	%
8.4	Ultimate Analysis.....	% by volume
	CO <sub>2</sub> .....	
	H <sub>2</sub> .....	
	O <sub>2</sub> .....	
	CO.....	
	CH <sub>4</sub> .....	
	C <sub>2</sub> H <sub>6</sub> .....	
	N <sub>2</sub> .....	
9.	Steady State Test.....	
9.1	Observed Output.....	Btu per hr
9.2	Fuel Burning Rate.....	Btu per hr
9.3	Air Delivery (Std Air).....	cfm
9.4	Efficiency.....	%
10.	Catalog Rating.....	
10.1	Output.....	Btu per hr
10.2	Fuel Burning Rate.....	Btu per hr
10.3	Air Delivery.....	cfm
10.4	Fan or Blower Motor Power (Name Plate).....	hp
10.5	Motor Voltage.....	volts
10.6	Amperage.....	amp

## Section C—Test Readings

Date of Test \_\_\_\_\_

11.	Testing Laboratory.....	
12.	Test Observer.....	
13.	Time of Reading*.....	
14.	Oil Fuel—Scale Reading.....	lb
15.	Oil Fuel Consumption, Avg.....	lb/hr
16.	Gas Meter Reading.....	cu ft
17.	Gas Consumption.....	cu ft/hr

\* Test sheet should provide space for 20 min. observations, 2-hr steady state test, and include column for total or average.

18.	Pressure.....	
18.1	Oil at Nozzle.....	psi
18.2	Gas at Meter.....	in.H <sub>2</sub> O
18.3	Gas at Manifold.....	in.H <sub>2</sub> O
18.4	Barometric.....	in.Hg
19.	Gas Temperature at Meter.....	F
20.	Gas Consumption (corrected to 60 F & 30 in. Hg).....	cu ft
21.	Flue Gas Analysis (by vol).....	%
21.1	CO <sub>2</sub> .....	%
21.2	O <sub>2</sub> .....	%
21.3	CO.....	%
22.	Flue Gas Temperature.....	F
22.1	Flue Gas Temperature Rise (Item 22 — Item 24).....	F
23.	Draft.....	in.H <sub>2</sub> O
23.1	Combustion Chamber.....	
23.2	Furnace Outlet.....	
24.	Air Temperature, Fan Inlet.....	F
25.	Air Temperature, Discharge Duct.....	F
26.	Fan Motor Performance.....	
26.1	Volts.....	
26.2	Amperes.....	
26.3	Energy Consumption.....	whr
26.4	Speed.....	rpm
27.	Fan Speed.....	rpm
28.	Oil Burner Motor Energy Consumption.....	whr
29.	Heat Exchanger Surface Temperature.....	F
29.1	Heat Exchanger Surface Temperature Rise (Item 29 — Item 24).....	F
30.	Casing Surface Temperature.....	
30.1	Casing Surface Temperature Rise (Item 30 — Item 24).....	F

## Section D—Computations and Results

31.	Heat Input, $H$ (Fuel Burned per hour $\times$ gross heating value).....	Btu/hr
31.1	For oil $H = \text{Item 15} \times \text{Item 7.2}$ .....	Btu/hr
31.2	For gas $H = \text{Item 20} \times \text{Item 8.1}$ .....	Btu/hr
32.	Heat Release per cu ft of Combustion Chamber Volume (Item 31 $\div$ Item 3).....	Btu/hr
33.	Fan Power Consumption (estimated) (Item 10.4) $\times \frac{\text{Item 26.1} \times \text{Item 26.2}}{\text{Item 10.5} \times \text{Item 10.6}}$ .....	hp
34.	Air Temperature Rise through Furnace (Item 25 — Item 24).....	deg
35.	Flue Gas Loss (Apply Items 21.1 and 22.1 to Fig. 1 for oil or to Fig. 2 for gas fuel).....	%
36.	Efficiency (100 — Item 35).....	%
37.	Gross Output (Item 31 $\times$ Item 36)/100.....	Btu/hr
38.	Air Delivery (Standard Air) $\frac{\text{Item 37}}{\text{Item 34} \times 1.08}$ .....	cfm

TABLE 2—CALORIFIC VALUES FOR FUEL OIL<sup>a</sup>

DEGREES API AT 60 F	DENSITY, LB/GAL	BTU/LB	BTU/GAL
24	7.587	19,190	145,600
25	7.538	19,230	145,000
26	7.490	19,270	144,300
27	7.443	19,310	143,700
28	7.396	19,350	143,100
29	7.350	19,380	142,500
30	7.305	19,420	141,800
31	7.260	19,450	141,200
32	7.215	19,490	140,600
33	7.171	19,520	140,000
34	7.128	19,560	139,400
35	7.085	19,590	138,800
36	7.043	19,620	138,200
37	7.011	19,650	137,600
38	6.960	19,680	137,000
39	6.920	19,720	136,400
40	6.879	19,750	135,800
41	6.839	19,780	135,200
42	6.799	19,810	134,700

<sup>a</sup> The above figures are from *National Bureau of Standards* Miscellaneous Publication M97 (Table 6).

TABLE 3—CORRECTION TO STANDARD API DEGREES AT 60 F<sup>a</sup>

OBSERVED TEMP OF OIL F	OBSERVED DEGREES API									
	24	25	26	27	28	29	30	31	32	33
50	24.6	25.6	26.6	27.6	28.7	29.7	30.7	31.7	32.7	33.7
60	24.0	25.0	26.0	27.0	28.0	29.0	30.0	31.0	32.0	33.0
70	23.4	24.4	25.4	26.4	27.3	28.3	29.3	30.3	31.3	32.3
80	22.8	23.8	24.8	25.7	26.7	27.7	28.7	29.6	30.6	31.6
90	22.2	23.2	24.2	25.1	26.1	25.1	28.0	29.0	30.0	30.9
100	21.6	22.6	23.6	24.5	25.5	26.5	27.4	28.4	29.3	30.3

OBSERVED TEMP OF OIL F	OBSERVED DEGREES API								
	34	35	36	37	38	39	40	41	42
50	34.7	35.7	36.7	37.7	38.8	39.8	40.8	41.8	42.8
60	34.0	35.0	36.0	37.0	38.0	39.0	40.0	41.0	42.0
70	33.3	34.3	35.3	36.2	37.2	38.2	39.2	40.2	41.2
80	32.6	33.6	34.6	35.5	36.5	37.5	38.4	39.4	40.4
90	31.9	32.9	33.8	34.8	35.8	36.7	37.7	38.7	39.6
100	31.3	32.2	33.2	34.1	35.1	36.1	37.0	37.9	38.9

<sup>a</sup> From *National Standard Petroleum Oil Tables*, Circular C410 *National Bureau of Standards*, March 4, 1936.

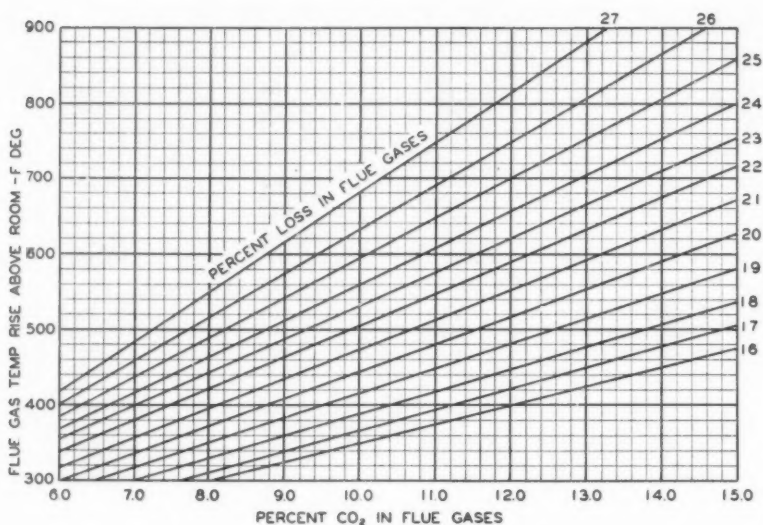


FIG. 1. FLUE GAS LOSS WITH OIL FUEL

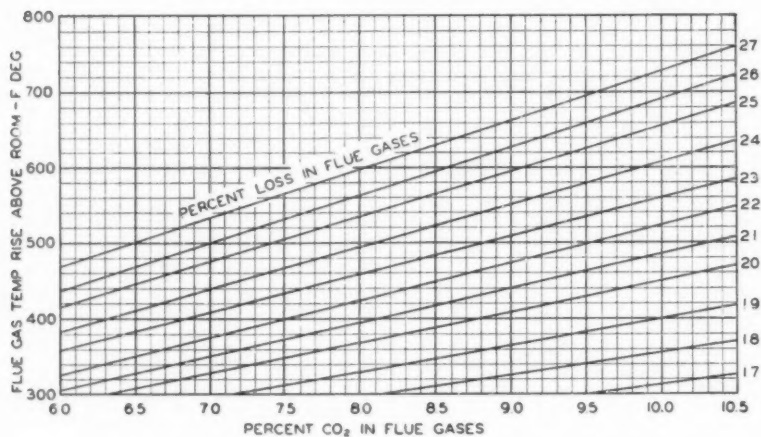


FIG. 2. FLUE GAS LOSS WITH NATURAL GAS FUEL

## In Memoriam 1955

NAME	JOINED
Roger W. Booth, Colorado Springs, Colo.	1949
Vernon R. Brown, Ft. Worth, Tex.	1952
James H. Carbone, New York, N. Y.	1937
Fulton W. Clare,* Atlanta, Ga.	1927
George H. Dahlquist, Chicago, Ill.	1953
Calvin R. Davis, St. Louis, Mo.	1927
Roderick I. Dennis, Toronto, Ont., Canada	1953
Edward L. Fox, Wyoming, Ill.	1948
Francis J. Fox, Minneapolis, Minn.	1945
W. J. Fritz, Seattle, Wash.	1952
Walter E. Gillham ( <i>Life Member and Treasurer 1926-29</i> ), Santa Barbara, Calif.	1917
Tony M. Ginn, Great Falls, Mont.	1935
Merlin J. Hauan, Seattle, Wash.	1933
George W. Hubbard ( <i>Life Member</i> ), Chicago, Ill.	1911
Charles H. Jackson ( <i>Life Member</i> ), Milwaukee, Wisc.	1923
David S. Jacobus ( <i>Life Member</i> ), Montclair, N. J.	1916
Edward B. Johnson ( <i>Life Member</i> ), New York, N. Y.	1919
John W. Jungels, Chicago, Ill.	1947
James C. Kennedy,* Syracuse, N. Y.	1945
W. A. Kopp, Phoenix, Ariz.	1949
Theodore E. Luzzi, New York, N. Y.	1944
Harry B. Matzen, West Palm Beach, Fla.	1940

\* Died in 1954 but Society not notified until 1955.

## In Memoriam 1955

(continued)

NAME	JOINED
James H. Milliken, Chicago, Ill.	1923
Leigh E. Nelson, Decatur, Ind.	1950
Vernon C. Page, Birmingham, Mich.	1936
George E. Perras, Montreal, Que., Canada	1936
Arthur L. Philastre, Kinston, N. C.	1950
Arthur L. Pistor, Montebellow, Calif.	1953
Philip R. Pullen, Rapid City, S. D.	1951
William J. Rapp, Indianapolis, Ind.	1953
Fred J. Scherger, Detroit, Mich.	1942
Claude B. Schneible, Detroit, Mich.	1950
William B. Schuler, Rock Island, Ill.	1947
William G. Seyfang, Buffalo, N. Y.	1939
C. B. Skinner, Jr., New Orleans, La.	1954
L. E. Slawson, Cleveland, Ohio	1938
Robert E. Stuff, Milwaukee, Wisc.	1949
Walter J. Temple, Kalamazoo, Mich.	1931
Leo A. Tilford, Jackson, Mich.	1941
Alf Tjersland ( <i>Life Member</i> ), Oslo, Norway	1906
Hill A. Tolerton, Canfield, Ohio	1944
Richard Y. C. Tom, Beloit, Wisc.	1954
Willard F. Uhl ( <i>Life Member</i> ), Minneapolis, Minn.	1918
Ephraim A. Vanderslice, Haddonfield, N. J.	1950
Paul J. Vincent, Baltimore, Md.	1931

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